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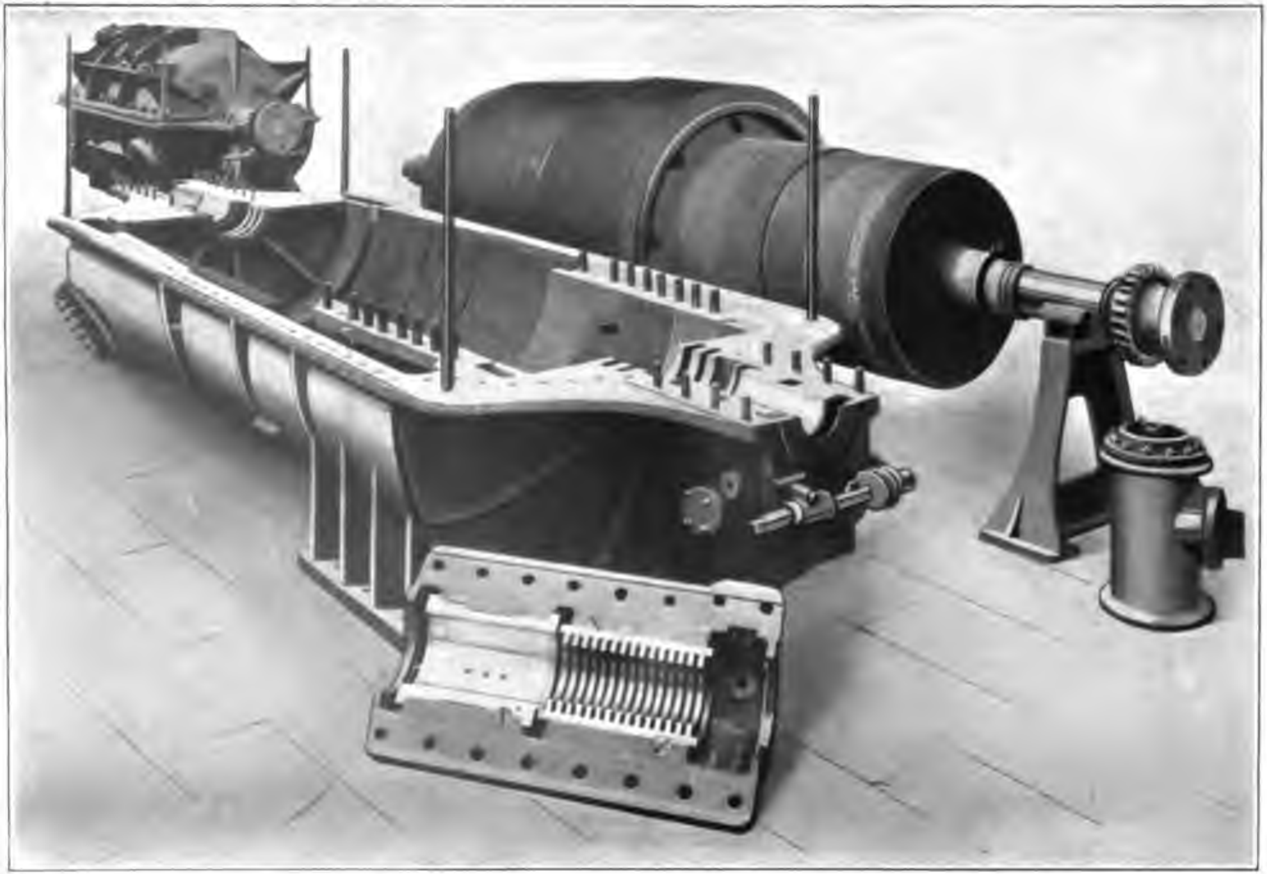

The High-pressure and One Low-pressure Turbine constructed by Messrs. A. & J. Inglis, Limited, Shipbuilders and Engineers, Glasgow, for the Khedive's Triple-Screw Yacht "Mahroussa."

A MANUAL

OF

MARINE ENGINEERING:

COMPRISING

THE DESIGN, CONSTRUCTION, AND WORKING OF MARINE MACHINERY.

BY

A. E. SEATON,

FORMERLY LECTURER ON MARINE ENGINEERING TO THE ROYAL NAVAL COLLEGE, GREENWICH;
MEMBER OF THE INSTITUTION OF CIVIL ENGINEERS; MEMBER OF THE INSTITUTION
OF NAVAL ARCHITECTS; MEMBER OF THE INSTITUTION
OF MECHANICAL ENGINEERS, ETC.

With Numerous Tables and Illustrations reduced from Working Drawings.

*SIXTEENTH EDITION, THOROUGHLY REVISED, ENLARGED,
AND IN PART RE-WRITTEN.*

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REESE

PREFACE TO SIXTEENTH EDITION.

THE demand for the Manual having exhausted the edition, the opportunity has been taken to make such alterations and additions as will bring it up to date. Although such a comparatively short time has elapsed since it was revised, the modifications and extensions of the Rules of the Board of Trade, Lloyd's Register, Bureau Veritas, British Corporation, &c., have been considerable, and they are now all noted in this edition.

The use of superheated steam on board ship has been revived in a tentative way; the advantages of steam in this state are obvious, and the possibilities and value of it are shown in diagrammatic form (Fig. 16*a*) from experiments made with engines of the marine type, or of the types analagous to it, in central electric stations on shore.

In Appendix N, the turbine as a marine motor is discussed.

In conclusion, the author trusts that this revised edition may meet with as kindly a reception as has been accorded to its predecessors.

A. E. SEATON.

WESTMINSTER, *March*, 1907.

196525

PREFACE TO FIFTEENTH EDITION.

THE First Edition of this Manual was written some twenty-five years ago when there was no similar book existing, when Lloyd's had no Rules for Machinery and the Board of Trade but a few elementary ones, when the Admiralty practically permitted each engine-builder to follow his own devices in design and his own rules for sizes, and when the British Corporation did not exist, nor very few of the Technical Schools now so freely scattered over the land. No better proof of the progress and change which has been made in this quarter of a century, I venture to think, can be found than by comparing this new edition with the original. I trust that this edition, which is virtually a new book, may be found to reflect the best engineering practice of the day, combined with the theories and discoveries of modern science, as the original Manual aimed to do, and, on the whole, succeeded in doing. I likewise hope that it may meet with the same kindly reception at the hands of engineers as was accorded its predecessors.

I have endeavoured to follow the plan originally observed, to make every point quite clear and free from ambiguity, and to give all rules and calculations their simplest expression without having recourse to the higher mathematics.

I have, in Chapter iii., given new methods for the guidance of engineers in determining the I.H.P. for certain speeds of ship in lieu of the old rules with so-called constants, from which the student had to make a selection without much to guide him in doing so; tables based on these new rules are given, which will save him much calculation in practice. The examples of steamships' performances are quite modern ones, and in every case taken from reports of carefully-conducted trials by eminent shipbuilders and engineers.

The original rules for screw propellers have been carefully tested by the best and latest experience. I am pleased to say that they have required very little readjustment. I have added to them, and elsewhere, many additional formulæ, which should be of great use in every-day engineering practice; for example, it is a long and tedious process to estimate the weight of a screw propeller; I have given a rule which will be found to give results sufficiently accurate for all practical purposes by which the weight may be obtained in a minute or two.

The chapter on Water-tube Boilers has been much enlarged and more fully illustrated. Other chapters have likewise been added to materially in the same manner. Three chapters towards the end are, moreover, entirely new—one on Oils and Lubricants, a most important subject seldom dealt with at all in engineering manuals; one on Weight of Machinery in all its Relations and Effects; and one on the Physical Effect of Weight and Inertia, in which the question of vibration and balancing of engines is fully dealt with.

I have very much extended the chapter on "Materials," and given as much information on metals as is possible in a work of this kind. I have further endeavoured, in Chapter xxiii., to give the student some guidance in questions where the endurance of materials is involved. I have, I believe, given due weight to the opinions, as well as to the experiments, of Wohler, Unwin, and others, without ignoring the results of every-day modern experience. I have also attempted to distinguish between the repetition of a positive stress, which I call *intermittent*, and that of positive and negative stresses, known as *alternating* stress. In marine engine design in the past, there has not been much evidence that the difference between these stresses and that due to a steady load was properly appreciated. In modern Admiralty designs, however, it is fully recognised.

For the very neat and original mathematical investigations involved and the methods of obtaining the information for balancing by weights, &c. (Chap. xxiv.), I am indebted to my friend and former pupil, Mr. A. A. Jude. I also take this opportunity of expressing my thanks to Mr. J. G. Dunlop, Clydebank; Dr. F. Elgar, of Fairfield; Mr. W. J. Pratten, Belfast; Mr. J. W. Reed, of Jarrow; Mr. E. W. de Russett, of Wallsend; and other friends, for much useful and valuable information, as well as photographs; and to Messrs. J. Stone & Co., for permitting me to include their "Table of Pressures on Screw Propellers."

Last, but by no means least, to my Publishers I would give my grateful thanks for the patience they have ever extended to me, for the care they have taken to make each edition of the "Manual" an improvement on its predecessor, and more especially now for the trouble they have taken with this one in embellishing it with illustrations and in preserving its handy thickness by enlarging the pages, although it involved the resetting throughout.

A. E. SEATON.

32 Victoria Street, Westminster, S.W.,
October, 1903.

ORIGINAL PREFACE.

THE following Work has been prepared to supply the existing want of a Manual showing the application of Theoretical Principles to the Design and Construction of Marine Machinery, as determined by the experience of leading engineers, and carried out in the most recent successful practice. The data on which it is based, now first thrown into form for publication, have been collected during many years of study and practical work. It is hoped that the volume will be found useful by the engineer and draughtsman engaged in practice as a Handbook of Reference, and by the student, launched for the first time on the intricacies of Marine Construction, as a guide, supplying to some extent his lack of experience.

The rules and formulæ introduced (which have been divested as far as possible of complexity, and given in the simplest form attainable) may be used by any one who designs with some regard to theory, and, by varying the constants, be made to suit his own ideas of strength and stiffness. It may, perhaps, be thought by some that in certain instances details have been entered into with unnecessary minuteness; but it should be remembered, on the other hand, that not every engineer has the contents of a well-filled drawing-office to fall back upon in cases of doubt and difficulty.

It is hardly necessary to premise that it is wholly impossible to reconcile the practice of the naval designer, who thinks more of efficiency and weight than of cost, with that of the mercantile engineer, who studies efficiency and cost with but small regard to weight, and, therefore, few rules can be given which shall absolutely suit both. However, the manufacturer of machinery for the Merchant Service might follow with advantage much that has been proved to be good in naval practice, and the Naval Authorities might again, on their part, borrow from the Mercantile Marine a few suggestions which would render a warship, while no less efficient than at present, perhaps somewhat less intricate for those who have to work her.

In conclusion, the author can but express a hope that the publication of these notes, imperfect as they necessarily are, may tend to make a little clearer some of the technicalities of Marine Design and Construction, and so help forward, in however slight a degree, the application of scientific investigation to those problems which the marine engineer is called upon, day by day, to solve.

HULL, *January 30th*, 1883.

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A MANUAL

OF

MARINE ENGINEERING.

CHAPTER I.

GENERAL INTRODUCTION.

THE first object aimed at by the marine engineer, is to propel a floating body through the water at a certain speed; the second, so to construct the propelling apparatus that the motion may readily be reversed; and the third, to adopt such an arrangement of propeller and engine as shall be convenient for the floating body and the service on which it is employed.

The principle on which nearly all marine propellers work is the projection of a mass of water in the direction opposite to that of the required motion. The only exception to this rule is the case of ferry steamers and some river craft, where a chain or rope lying in the bed of the river passes over a wheel or barrel in the ship itself.

The water, in modern practice, is projected by—(1) One or more screws at the ends of the ship (as will be described under the heading of *propellers*); (2) by one or more paddle-wheels outside of the ship; or, (3) by a form of wheel in the inside of the ship, which is generally spoken of as a *jet*-propeller, as the water issues in jets from orifices in the ship's side.

The Paddle-Wheel.—The oldest of these three forms is the paddle-wheel; and although the screw-propeller has almost entirely superseded it in sea-going ships, it still maintains its position in river steamers, and in some cross-channel steamers of large power and light draught of water. Inasmuch as the screw, to work efficiently, must be wholly submerged, and any increase in its size demands an increase in the draught of water, whereas the paddle-wheel can be increased by making it broader on the face without increasing the *dip* of the floats: it follows, that where large power is required with a light draught of water, the screw or screws cannot be adopted, unless the diameter is less than the limit of draught,* and recourse must be had to paddle-wheels. Beyond this, the few claims which paddle-wheels have are limited to special cases, and even then their merits—such as decreased vibration, handiness in tugs, and steadiness in a seaway—are only comparative; while against these are some very definite objections—viz., great weight, the amount of space occupied (and that in the most valuable part of the ship), and the liability of the wheels to damage, together with the great increase in breadth of ship, and the increase in the surfaces exposed to sea and wind.

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* By a *special formation* of the stern, light draught ships can be propelled by screws in smooth water with fair efficiency when the diameter exceeds the draught.

The Screw.—The screw, on the other hand, is much smaller for equal powers, is wholly immersed, and also protected by the quarters of the ship; the propelling force is applied to the strongest part of the ship; the piston speed being much higher than that of the paddle-wheel engine, the engines are both smaller and lighter, and consequently cheaper, and the straining action on the hull is much less; the engines, lastly, can be placed in any convenient position, and occupy less space than those of the paddle-wheel. With these advantages there is the drawback that the screw, when badly designed, causes vibration of the stern of the ship, and with large powers and single screws produces severe side strains, which the form of stern is not calculated to resist well; and, again, the screw itself is apt to foul rope and chains, while, from its liability to *race*, it is not such an efficient propeller against a strong head-wind and sea as the paddle-wheel.

The Jet-Propeller.—The *jet*-propeller, the invention of Mr. Ruthven and many others, has not been received with general favour, and has had accorded to it very few practical trials. In this case the propeller is inside the ship, in the shape of a turbine wheel, or centrifugal pump, and so has the advantage of being wholly protected, while no considerations as to draught of water determine its adoption or rejection. It occupies a great amount of space in the ship, and requires large openings in the skin for inlets and outlets, and should the section of the stream of water be reduced by increasing the velocity, one of the great objections to the system—viz., the friction in the pipes and passages—is very much aggravated.

Reversing.—The reversing of the motion of the ship is obtained by reversing the motion of the propeller, and in this respect the paddle-wheel is more efficient than the screw, since the floats leave the water at nearly the same angle as they enter, and also present flat surfaces to the water; further, the stream of water from the paddle is not projected against the ship as it is from a screw, when going *astern*. The features in a screw which cause it to work efficiently and without vibration when going ahead, tend to produce the opposite results when its motion is reversed.

A consideration, therefore, of the qualities of each form of propeller, and of the necessities of the service on which the ship is to be employed, must determine which particular form shall be adopted.

Different Forms of Engine.—The third consideration for the engineer, as has been stated, embraces the machinery as well as the propeller; and since each form of propeller may be driven by engines of different design, each design has its special features, which will be found suitable or the reverse for special services, and which, where the question is an open one, will either commend or condemn it according to the practice or the prejudices of the engineer.

The Beam Engine.—Naturally the beam engine was the first which commended itself to the minds of the pioneers of steam navigation, since, at the time when the propelling of ships by steam was only emerging from its embryo state, that particular form of steam-engine was the only one which had been proved by experience to be a success. Hence, we find Bell's "Comet" with a modification of a beam engine (fig. 1), since improved upon, under the name of the *Side-Lever* Engine; this reached its highest pitch of perfection in the hands of the Napiers on the ships of the celebrated Cunard Line of Atlantic Steamers, as shown in fig. 2. This class of engine still exists in two forms, the one, as in fig. 2, where the fulcrum or weigh-shaft is at or near the centre of the beam, the connecting-rod being at one end and the side rods from the piston-rod crossheads at the other; this is the true "Side-Lever" Engine. Fig. 3 is an interesting example of an early engine worked by means of a beam of the "bell crank" type, made by Maudslay and fitted in a Thames passenger steamer in 1816. In this engine the connecting-rod is nearly horizontal, so that its weight and that of the crank would always keep the latter off the "dead" points. When the fulcrum or weigh-shaft is at one end, as was the case in the engines of the "Comet," it is usually called a "Grasshopper" Engine, and in this form was much used in tug-boats and river steamers. The chief advantages possessed by this engine, as shown in fig. 4, are cheapness of construction, consequent on simplicity of design; a long stroke of piston can be obtained in a shallow ship; the racking action from the engine when in motion is very slight, and taken by the strongest part of the ship; and when only a single cylinder is employed, there is in practice no "dead" point—that is, the crank can be moved from any position in which it may have stopped. This latter quality is due to the position of the connecting-rod with respect to the levers when the piston is at the end of its stroke, and the slight amount of *play* in the brasses. This class of engine is capable of working satisfactorily when in such a bad state of repair that the same condition in any other form of engine would prove dangerous; moreover, it seems to require far less care and attention than are usually found necessary with marine machinery. The objections to it are that it

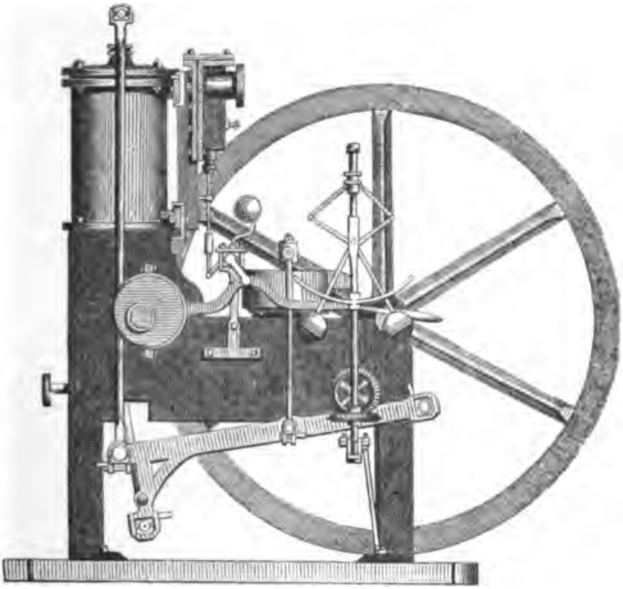

Fig. 1.—Engine of the "Comet," 1811-12.

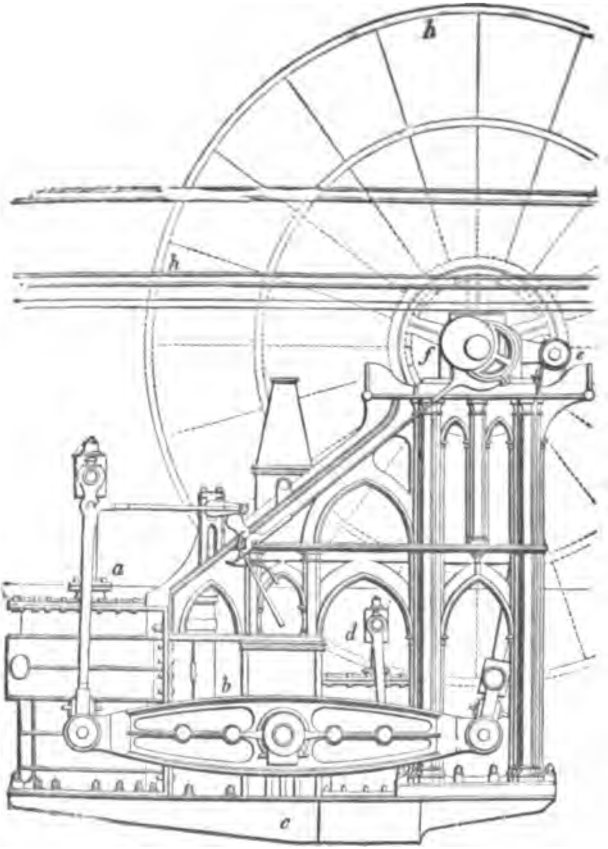

Fig. 2.—"Side-Lever" Engine.

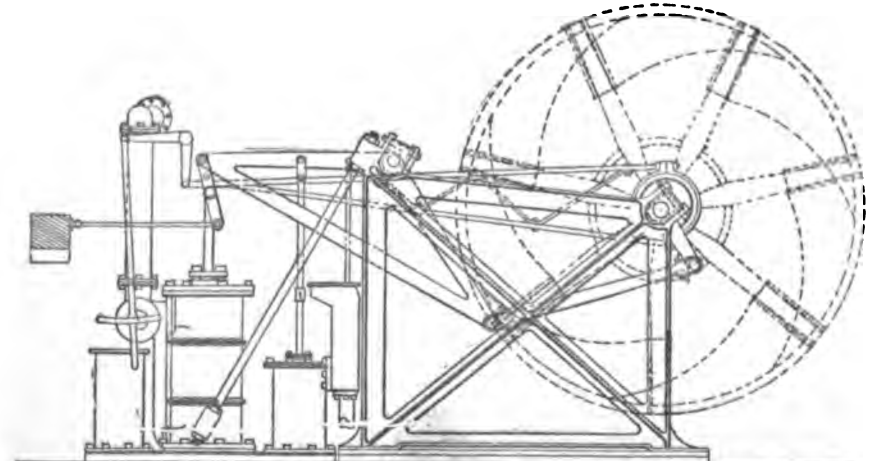

Fig. 3.—Engines of Thames Steamer "Regent," 1816 (*Maudslay*).

is somewhat heavier, and takes up more room in the ship than more modern forms.

The beam engine, in its earlier and original form, is still adopted by American engineers, both in river and coasting steamers, and in their hands it has proved a success; but the objections to it are serious for sea-going ships, especially those of European form, which have not so great a height of deck above the water-line.

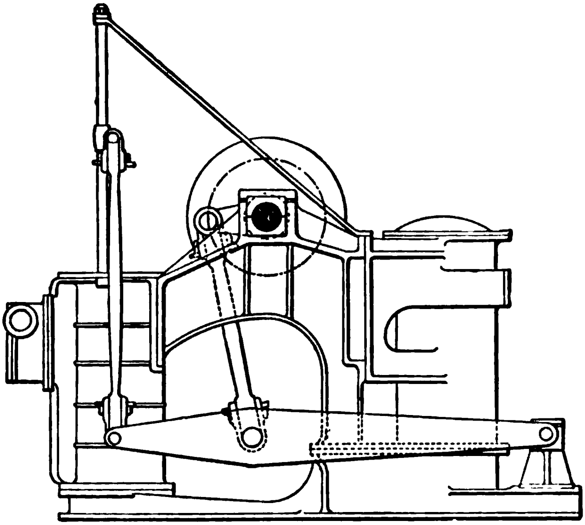

Fig. 4.—"Grasshopper" Engine.

The Steeple Engine (fig. 5) is one of the earliest forms of marine engine where the piston operated directly on the crank. It was a favourite engine on the Clyde, and possesses the advantage of taking up little room, is moderately light and cheap, and has fewer working parts than the side-lever engine; but, on the other hand, the length of stroke is limited by the depth of the ship; moreover, considerable height is required above the shaft. In quite recent times Laird Brothers have adopted this type of engine for paddle steamers of large power.

The Oscillating Engine (represented in figs. 6 and 6*a*), first suggested by Trevithick, and brought to its highest pitch of perfection, first by Messrs. Maudslay & Field, and then by Messrs. John Penn & Sons, is, on the whole, one of the best adapted for paddle-wheels under ordinary circumstances. It is the lightest and most compact form, and has fewest working parts. It can be arranged with the cylinder vertical in its mean position, or inclined even as far as the horizontal position; it will work best, of course, in the former position, and the chief objection to its more frequent adoption in shallow draught steamers is the want of room vertically to effect this. It is not so convenient a form when steam of higher pressure is employed, from the difficulty of getting an efficient apparatus for an early cut-off of steam, and the liability of the trunnion to leak; the former is the real difficulty, the latter existing more in imagination than practice, and the solution has been found in the introduction of compound cylinders. The most successful oscillating engines, however, have been those of large power, working with a steam pressure not exceeding 30 lbs. per square inch, and consuming on the average 2¾ lbs. of coal per I.H.P. per hour; and as they have been, and are likely to be, only fitted in ships where consumption of coal is not of first importance (as in fast passenger steamers on short voyages, and in yachts on similar service), it seems wise to avoid the higher pressures, especially as with them the cost and weight of machinery is materially increased. The inclined position of cylinder has been adopted in

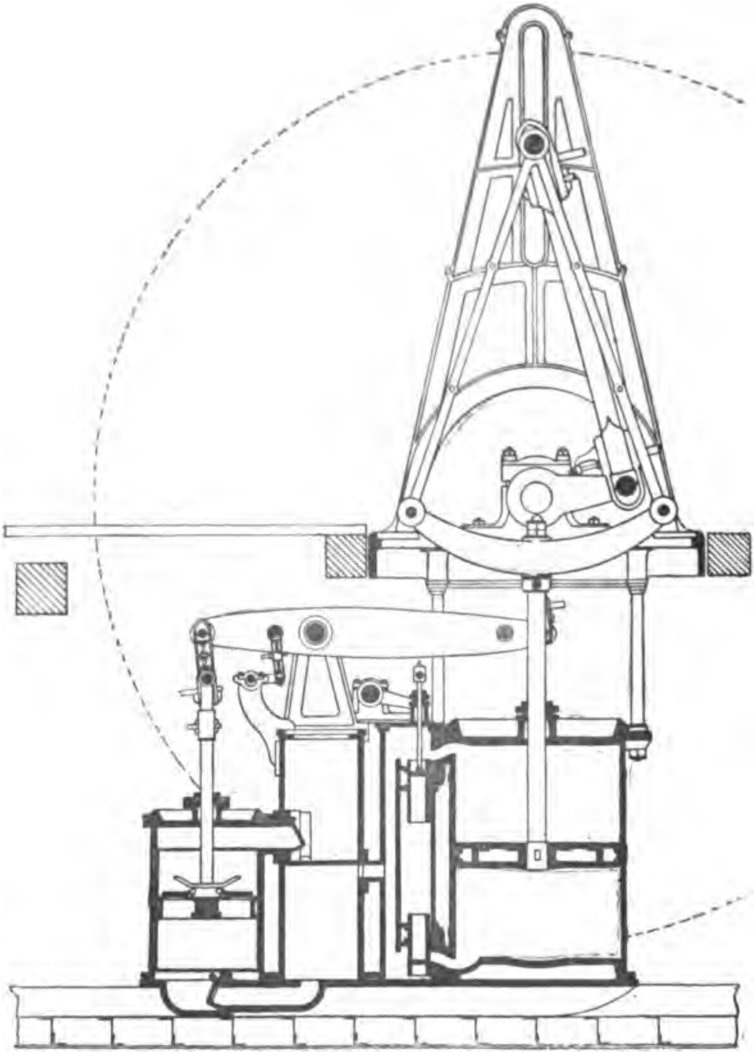

Fig. 5.—Steeple Engine.

fast, shallow draught steamers and naval tug boats with success; and to get over the difficulty caused by the large space athwart-ships required by this type of engine with two cylinders, one cylinder has been placed before the shaft and the other abaft it, both operating on the same crank.

The Diagonal or Inclined Direct-acting Engine, the invention of Mark Isambard Brunel, in 1823, is one that, although not therefore new, was not so much adopted in former years as it has been latterly. It is simply a horizontal engine, set at an angle so as to suit the height of the shaft at one end and the frames of the ship at the other. It takes up a large amount of space in a fore and aft direction, but not so much in the athwart-ship direction as either the oscillating or side-lever type, and for this reason no doubt, and because it is somewhat heavier and more expensive than the

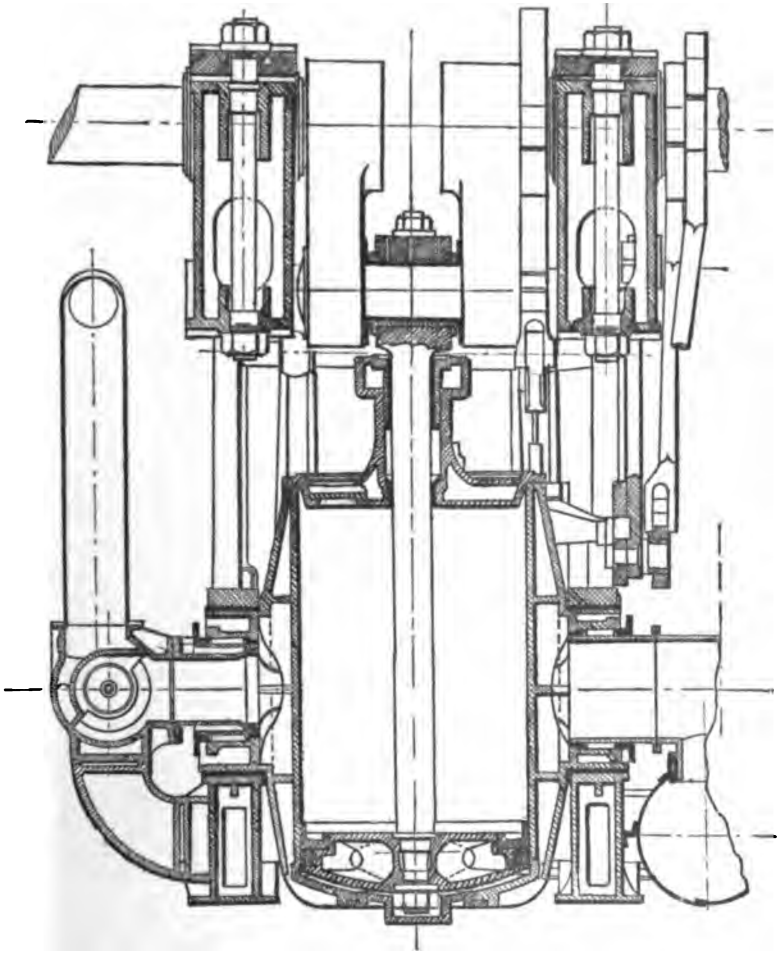

Fig. 6.—Oscillating Engine—Section through Trunnions.

other forms of paddle engine, it had not found so much favour with engineers. It is, however, a very convenient form when space is not an object in design, especially with large powers in light draught vessels, as the weight is not so much concentrated, and the strains are in a direction where the natural structure of such ships is best calculated to resist them. For these reasons this form of engine is now adopted generally in the fast river and cross-channel steamers. In some ships of very light draught, the heavy cast-iron framing is replaced with light wrought-iron, or steel structures, sometimes designed to form part of the framework of the ship; when this has been done the weight of the engine has not exceeded that of other types. Fig. 7 is a good example of a modern diagonal engine as fitted in fast, shallow draught vessels by Messrs. John Brown & Co., Clydebank, and others. Fig. 8 is an example by the Fairfield Company of a large

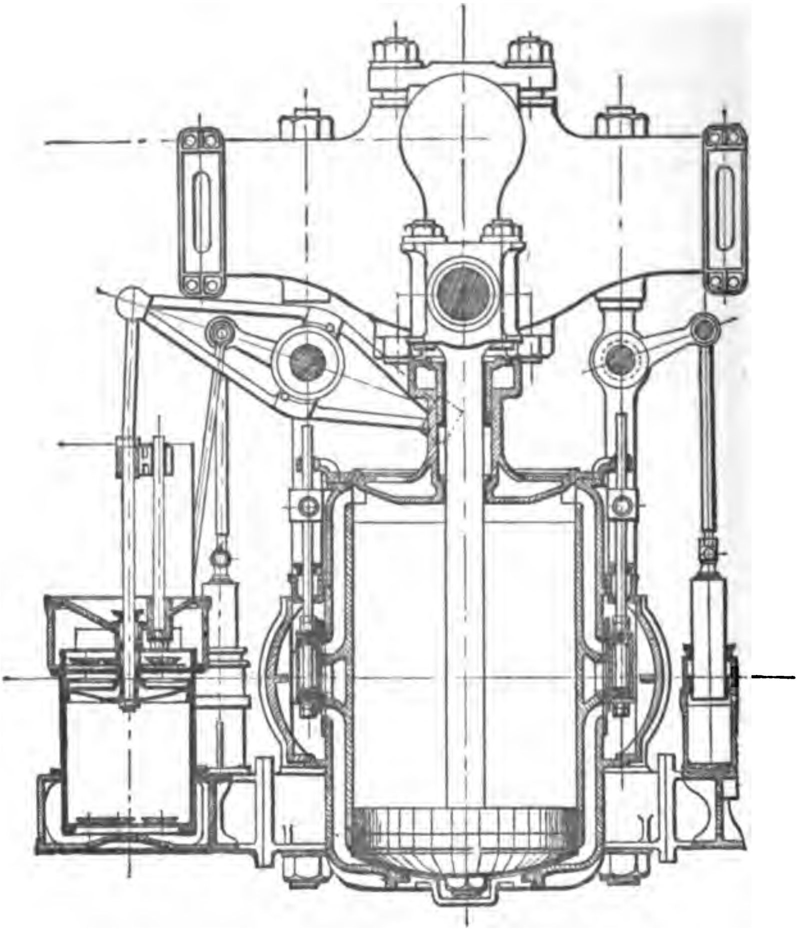

Fig. 6a.—Oscillating Engine—Section through Valve Boxes.

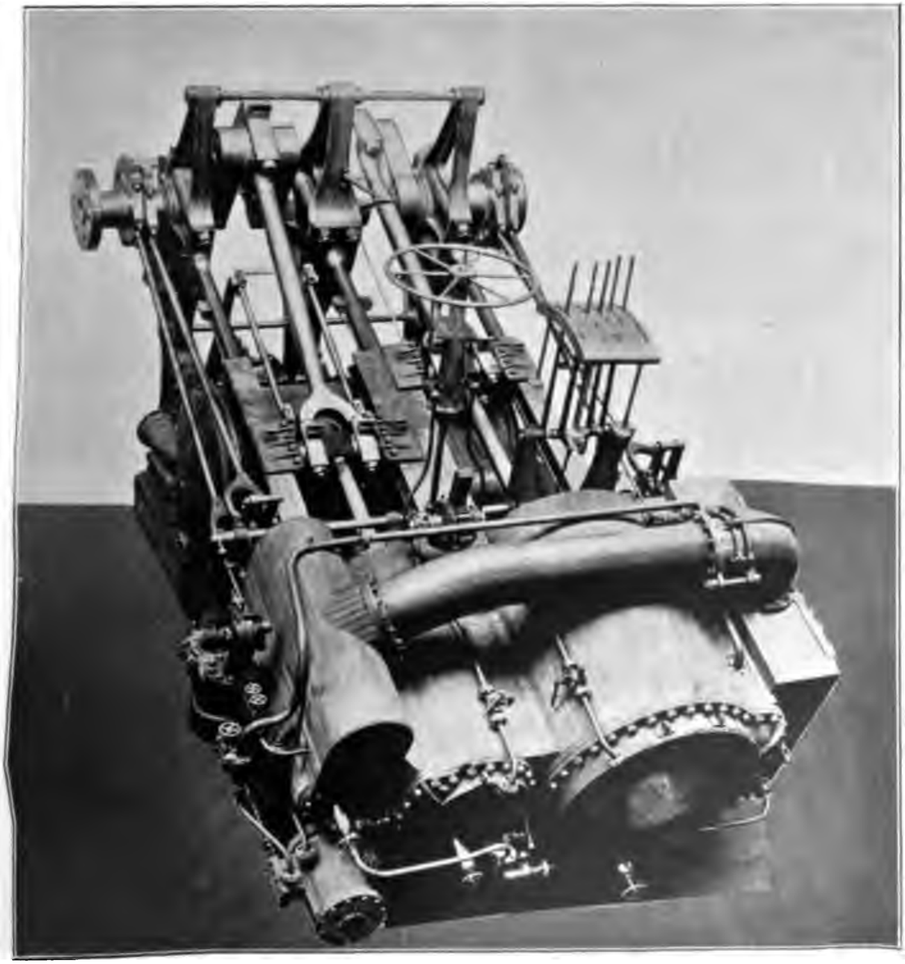

Fig. 7.—Diagonal Compound Paddle Engines.
(J. Brown & Co., Clydebank.)

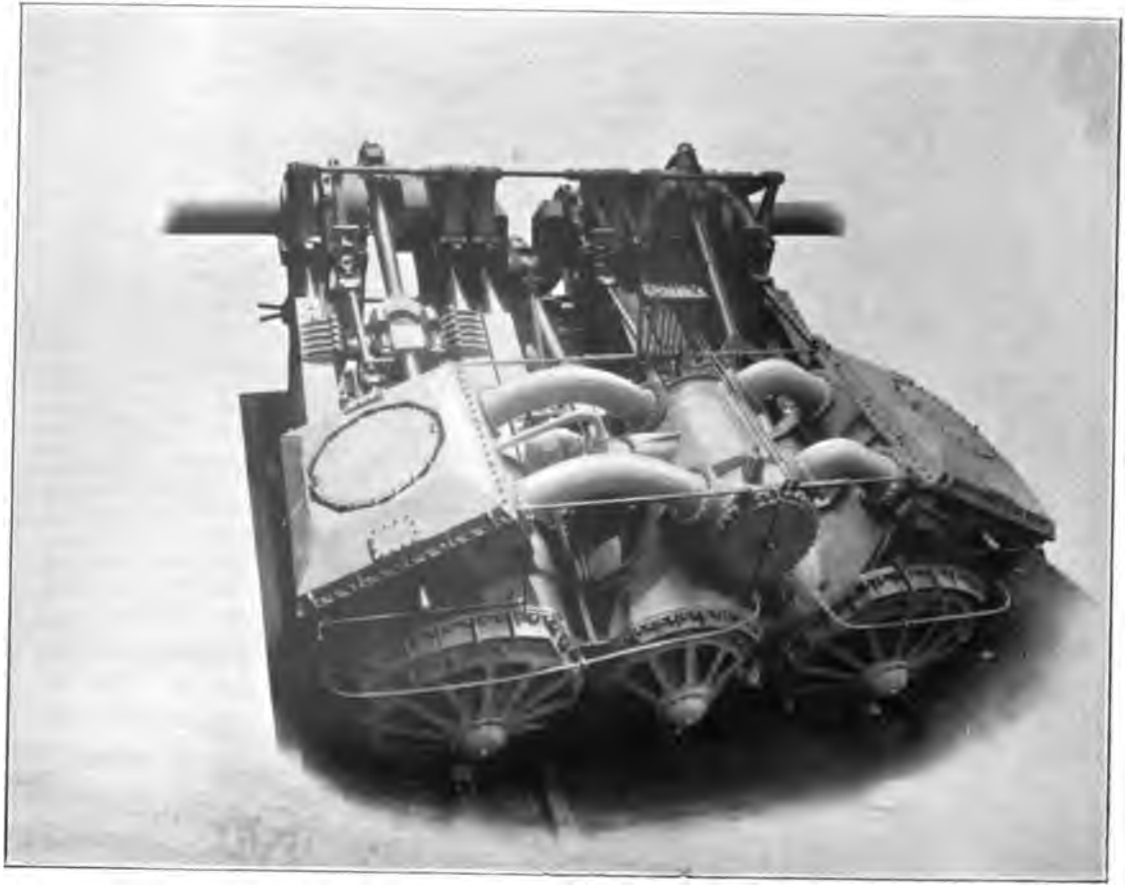

Fig. 8.—Three-crank Compound Diagonal Paddle Engines.
("Empress Queen," 10,500 I.H.P., Fairfield Co., Govan.)

diagonal engine where two low-pressure cylinders are necessary, and consequently three cranks are required. This is the most powerful paddle engine ever made.

Triple-expansion engines with three cranks have also been made to a limited extent.

Single-cylinder diagonal engines, with expansion gear to permit of high-pressure steam being used, are quite common in the West of Scotland.

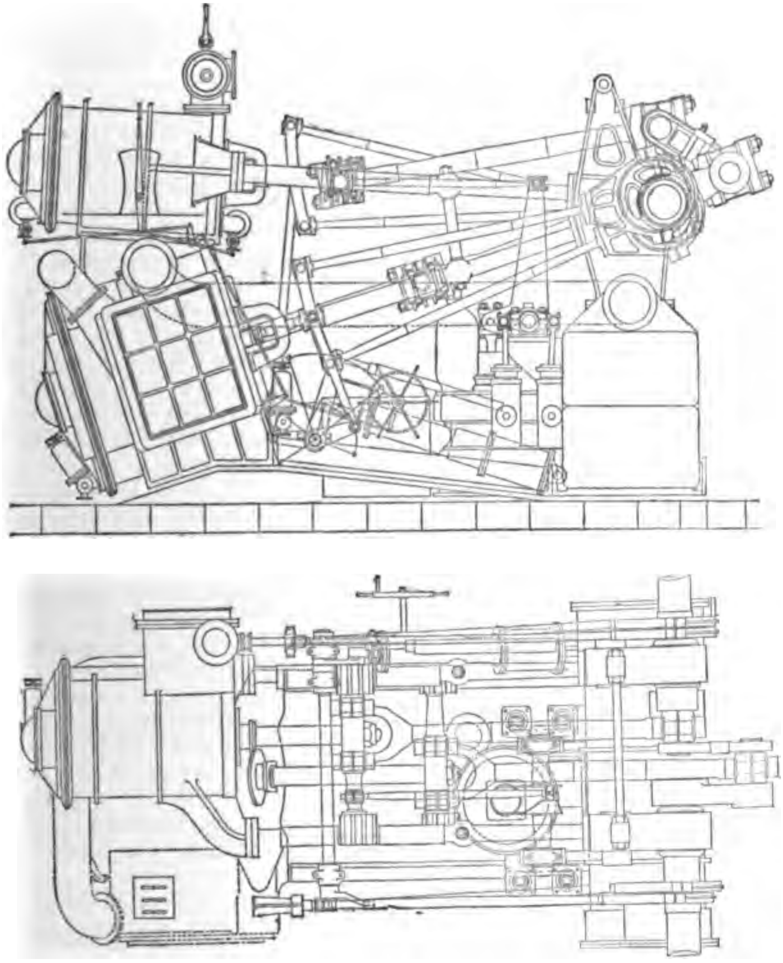

Fig. 9.—Diagonal Compound Paddle Engines.

They are, of course, cheap, and suit the comparatively narrow ships in which they are fitted; moreover, they are easily handled, as the weight of the crank and connecting-rod keeps them off the "dead" points. The momentum of their moving parts, however, sets up an action on the ship which is by no means pleasant to the passenger. The equivalent of a two-crank engine is obtained (fig. 9) by placing the high-pressure cylinder above the low-pressure cylinder of a diagonal engine in a horizontal or nearly horizontal position, connecting it by columns to the main bearing frames in the same way as the low-pressure cylinder is done, and connecting the piston-rod crosshead to a pin on the crank-arm opposite that whose pin is connected to the low-pressure crosshead; the two cranks are then set at an angle to one another equal to 90°, less the angle between the centre lines of cylinders, and connected by a drag link. This form of engine is a very convenient one where athwartship space is of consequence and a compound engine is required; it is also cheaper and lighter than the engine shown in fig. 7. Fig. 9 is an example of such an engine.

Other Forms of Engines.—There have been many other forms of paddle-wheel engines, such as the "Gorgon" Engine, a form of direct-acting engine; the "Annular Cylinder" Engine of Messrs. Maudslay, in which the connecting-rod was arranged so as to be partly within a cylinder placed in the centre of the steam cylinder, so that the piston was *annular;* the twin cylinder engine (fig. 10) of the same eminent firm, in which the same idea as to the connecting-rod was carried out; the open cylinder or atmospheric engine, in which steam was admitted only to the underside of the piston; the trunk engine, in which the piston had a trunk or cylinder on its top side instead of a rod, and in which the connecting-rod worked; and others, all of which have ceased to be of interest except from historic considerations, as the types previously particularised, and which will be more fully described later on, have entirely replaced them.

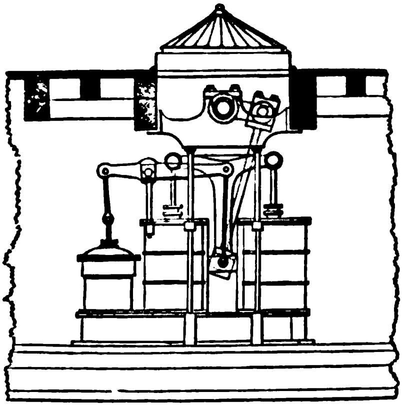
Fig. 10.—Twin Cylinder Engine.

Screw Engines.—On the introduction of the screw as a means of propelling ships, it was only natural that engineers should seek to add to the then existing forms of marine engines to drive it, rather than to invent new ones, as the fate of the screw for many years might have otherwise been that of the engines—viz., consignment to the dark confines of the Patent Museum. As the screw required to be run at many more revolutions per minute than the paddle-wheels, and there had been established definite limits for the number of revolutions of the engines, it became necessary to introduce wheel-gearing to satisfy these two conditions. But as an example of the manner in which conditions may continue to impose themselves long after their necessity has ceased to exist, gearing remained as part of a screw engine, with types of engines which might safely and easily have been run at the same number of revolutions as the screw, or rather, at the number of revolutions a screw *might* have been run at, if engineers had made their screws sufficiently large; for cause and effect had so changed places, that small screws existed long after the time when engineers had found how to make large ones of sufficient strength. So, beam engines, side-lever engines, oscillating engines, direct-acting engines, and many of the other forms were made to drive the screw by the intervention of cog-wheels, and for many

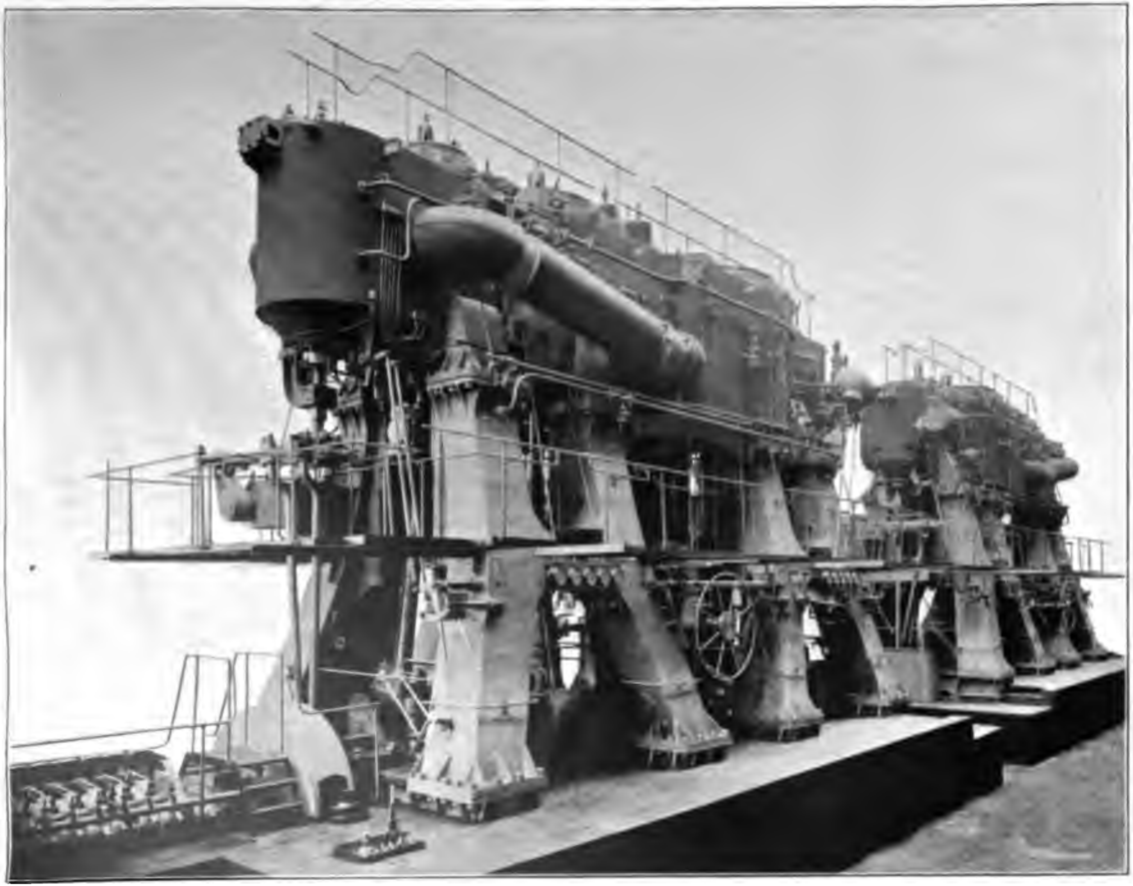

Fig. 10*a*.—Vertical Quadruple Expansion Engines—T.S.S. "Kinfauns Castle" and "Kildonan Castle," 10,000 I.H.P., Fairfield Co., Govan. Cylinders 28, 39¾, 57½ and 82 inches in diameter × 5 feet stroke.

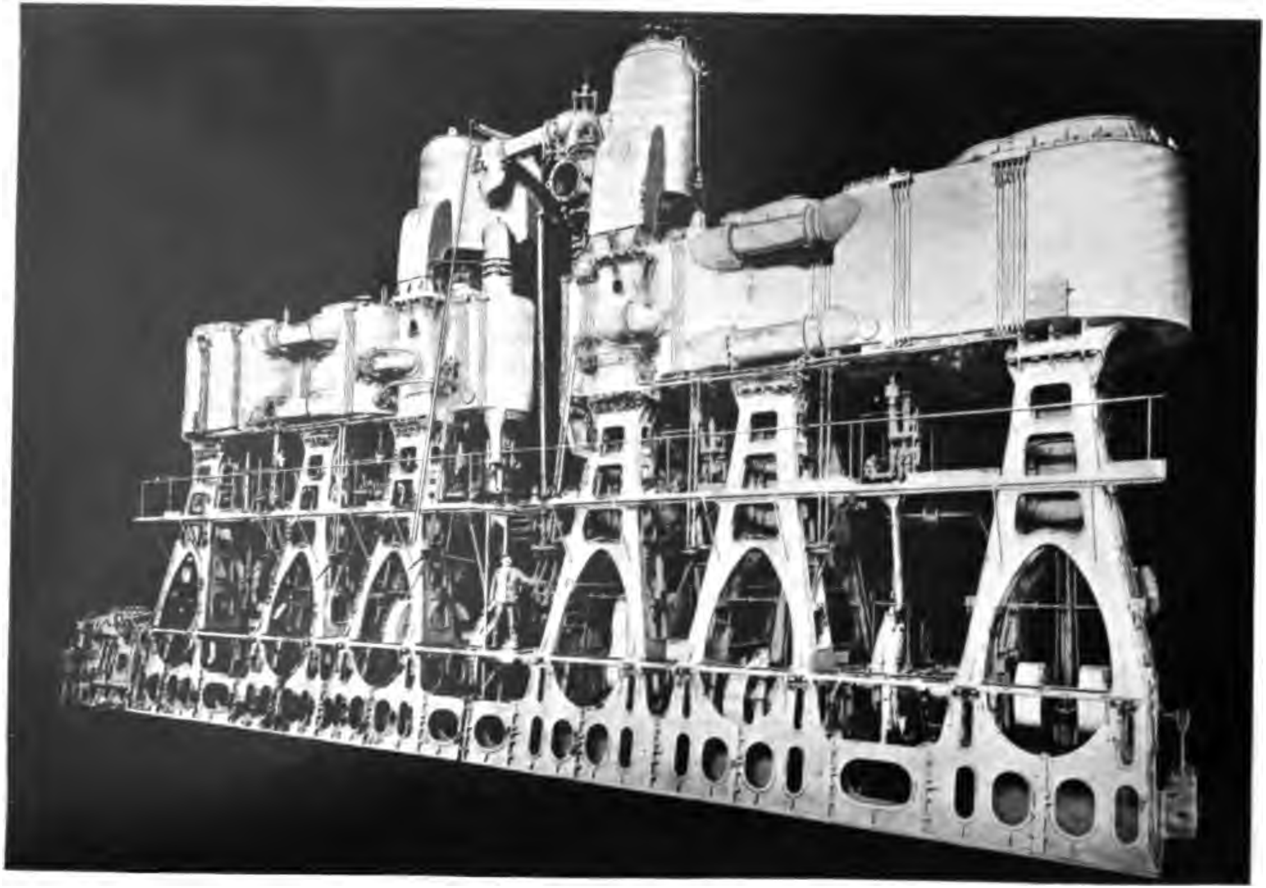

Fig. 11.—Quadruple 8-Cylinder Engines of T.S.S. "Kaiser Wilhelm II.," 40,000 I.H.P., Cylinders 112·2, 74·8, $\frac{37\cdot4}{49\cdot2}$, $\frac{37\cdot4}{49\cdot2}$, 74·8, 112·2 inches diameter × 70·8 inches stroke. "Made in Germany."

years the millwright was an important functionary in a marine-engine shop, especially where repairs were executed. No doubt, it is owing to these considerations that the paddle-wheel so long maintained its position; the machinery of a screw ship was as heavy as that of a paddle-wheel ship; it occupied more space, and the wear and tear was much heavier, while the belief in the power of the visible wheel was far more sincere than that in the invisible screw, and it required the experiment of the "Alecto" and "Rattler" to convince wavering minds. It is not necessary here to go into any detail as to the early forms of screw engine, as they are now extinct, and the interest attaching to them is purely historical. It is, therefore, only to show the genesis of the marine engine, and to account for certain "survivals," which will be noted further on, that the subject has been entered upon here at all.

The first step from the geared engine was to adopt modified forms of the paddle engine, so as to work direct on the screw shaft. The oscillating cylinder was employed vertically by inclining the shafting so as to be high enough at the forward end for the cylinder to work under it; then the cylinder or cylinders were placed diagonally, and finally inverted vertically over the shaft, which was inclined the reverse way to that described above. Maudslay's annular cylinders were inverted above the shaft. Beam engines even were arranged to work direct; but with very limited success. The disc engine originally invented by Hornblower in 1804, and improved by Bischop, and the pendulum engine of Captain Ericsson, were also used for driving the screw shaft directly, as were also various forms of rotary engines. Finally, the direct-acting engine was changed from the diagonal or inclined position to the inverted vertical. Messrs. J. & G. Thomson, Glasgow, were the first engineers to adopt this form; they were, however, soon followed by others. It has ever since been the most approved and successful engine for merchant steamers, and is now adopted by the British Admiralty for all classes of warships.

Another engine which soon attracted the attention of engineers seeking a design for direct-working screw engines, was the "Steeple" form; for, when laid horizontally and modified to suit that position, it made a useful arrangement, especially for warships and others requiring the machinery to be low down in the hull. This became known then as the "Return Connecting-Rod" Engine, and was for many years very generally adopted by engineers, sometimes from necessity, but oftener from choice. It is still in very general use in the older warships. Direct-acting engines, also, were adopted in a horizontal or nearly horizontal position by some engineers for warships. These, too, have held their own, and have many advantages over all other horizontal forms, especially in later days, since the introduction of higher pressures of steam; and they are likely to hold the same position in the estimation of engineers for warships, when required for some special reason to be horizontal.

The late Mr. John Penn invented and introduced the horizontal "Trunk" Engine, and its performance in the frigate "Arrogant," of 900 I.H.P., was so satisfactory, that the Admiralty adopted it very extensively, and continued to use it even for compound engines. It was taken up by a few of the larger steamship companies, but was finally abandoned by them in favour of vertical engines.

Various attempts have been made from time to time to drive the screw by what are called "Rotatory" Engines—that is, engines in which a rotatory motion is got direct from the piston or its substitute without the intervention of rods in the ordinary way; but although some of them had shown great ingenuity on the part of their inventors, and had, in some cases, worked satisfactorily for a time, none of them had sufficient success to enable their promoters to go beyond the experimental stage (*vide* Rankine on *The Steam Engine*, pp. 503-4). The interest in them has, however, been revived by the great success achieved by Mr. Charles Parsons with his steam motors in the s.s. "Turbinia" in 1897, whereby a speed of 34 knots per hour was attained, and, later, in the destroyers "Viper" and "Cobra" (both unfortunately lost when new), and in the passenger steamers "King Edward" and "Alexandra" and other similar ships, which have run successfully at speeds from 19 to 23 knots, and H.M.S. "Amethyst" 23·6 knots. The Curtis & Rateau turbines have been also successfully adopted for marine work.

Of the screw engines above sketched out, and now existing as the types of modern engineering practice, the direct-acting vertical form deserves first attention, as it is the only one which is now employed in both merchant ships and warships.

The Vertical Direct-acting Engine consists essentially of one or more cylinders, supported on columns, and having piston-rods, guided on suitable slides, generally attached to the columns; and the lineal motion of the piston converted to rotary motion by means of a connecting-rod from each piston-rod end to the crank-pin. This arrangement of engine admits of longer stroke of piston than obtains in the horizontal engines: the working parts are well above the bilge of the ship, and in clear view of the engineer; the weight of the pistons is taken on the crank-pins, so that the energy stored in them in the upstroke is given out on the downstroke, instead of being on the side of the cylinders, as in horizontal engines, when it only serves to cause frictional resistance to their motion; the same comparisons hold good for the valves, piston-rods, and connecting-rods. There is not so much racking strain on the structure of the ship, as with horizontal engines; and there is greater scope for variety of design and arrangement of detail with this class than in almost any other. On the other hand, their height causes the centre of gravity of machinery to be somewhat higher than that of horizontal engines, and also generally prevents their use when it is a necessity for the machinery to be below the water-line, as with unprotected warships, or when the 'tween decks are required as clear as possible of obstruction. This latter, however, was an argument used only by the opponents of vertical engines, as, whatever engine is fitted in a ship, there must be hatches, &c., immediately over it for light and ventilation, which hatches cannot be much smaller than would be necessary for a vertical engine. The vertical engine requires less attention when at work than the horizontal, and the wear and tear is considerably less. Under these circumstances, it is not astonishing that it entirely replaced the horizontal engine in the mercantile marine, and has done the same in the Navy.

The Horizontal Direct-Acting Engine works on precisely the same principle, and has the same essential working parts as the vertical before described; but inasmuch as only half the beam of a single screw ship is available for those parts, the stroke of the piston was exceedingly short in ordinary sized warships, whose beam far exceeds that of large merchant steamers, and was not very long in even the largest warship.

Since, with a fixed number of revolutions, a certain capacity of cylinder

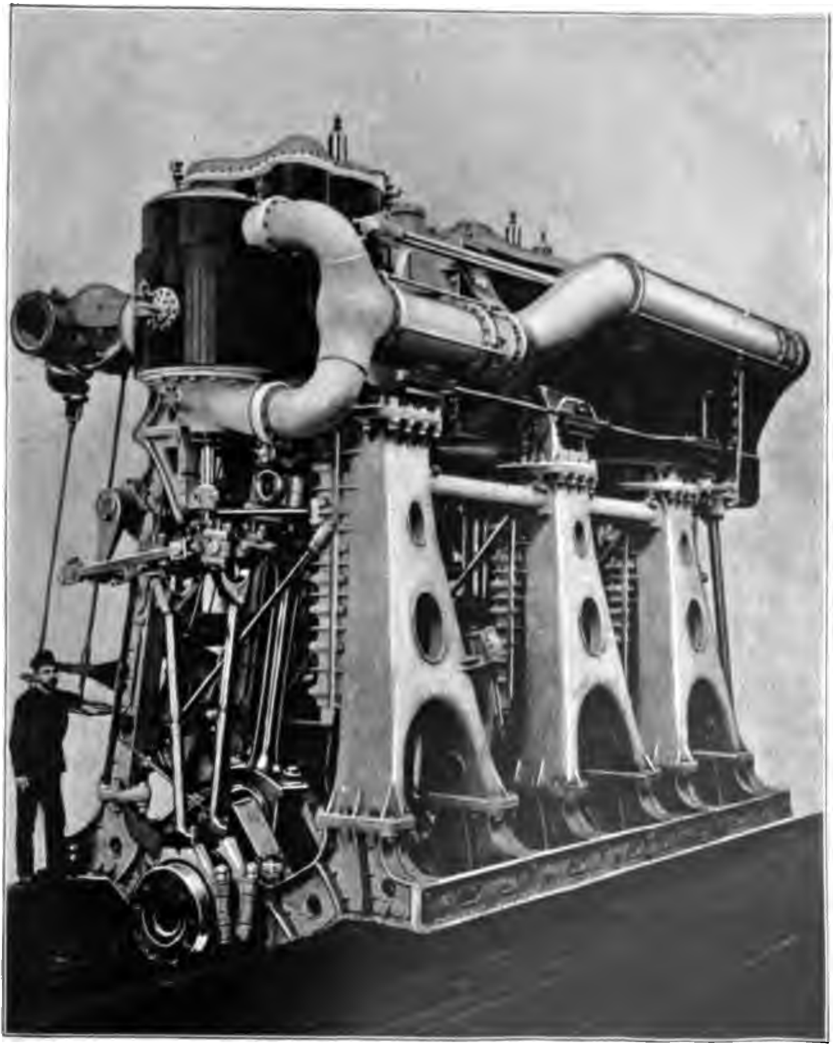

Fig. 12.—Triple Compound Engines of H.M.S. "St. George," 12,000 I.H.P., Cylinders 40, 59, and 88 inches diameter × 51 inches stroke. Designed by the Author.

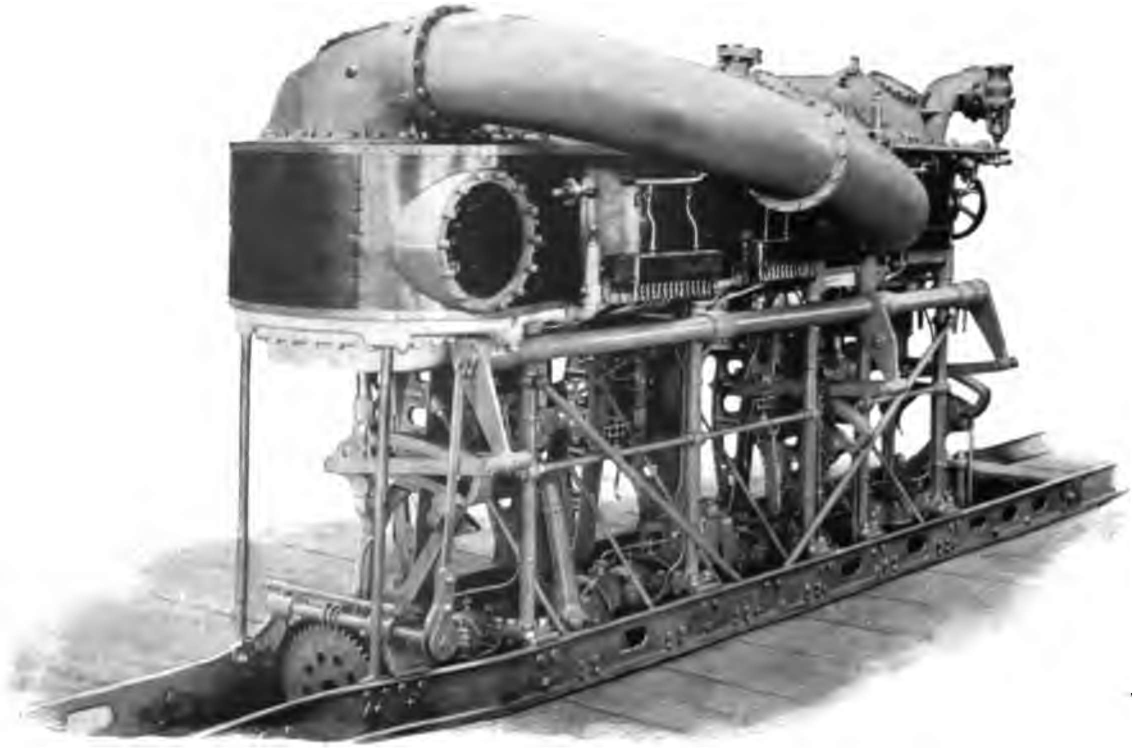

Fig. 13.—Engines of H.M.S. "Salmon" (Destroyer), 4000 I.H.P. Designed by the Author.

is necessary for a certain power, it follows that any diminution in the length of the cylinder necessitates a corresponding increase in its diameter; hence, these very short stroke engines have abnormally large pistons for their power, and, consequently, an aggravation of the evil arising from the weight of the pistons. As the momentum of the pistons had to be balanced by fitting heavy balance weights opposite the cranks, to reduce the racking on the framework, both of engines and ship, and as these weights cannot conveniently be placed very far out from the centre of the shaft, this class of engine was more difficult to balance than others of the horizontal type.

It may be said, however, in their favour, that they showed themselves in practice to be efficient in working, and, on the introduction of the compound engine and higher pressures of steam, they found greater favour than in the days when, for reasons stated below, 30 lbs. pressure was considered the highest safe limit for large engines. They possess the advantage of having their working parts well in view, and easy of examination and repair.

Penn's Trunk Engine (figs. 14 and 14*a*) differed from all others by having no piston-rod as an intermediary between the piston and the crank-pin, for the connecting-rod hinges on a pin or *gudgeon* in the centre of the piston, and was surrounded by a cylindrical case or *trunk* concentric with the cylinder, attached to the piston, and passing through a stuffing-box in the front of the cylinder; there was also a similar trunk in rear of the piston, which served as a support for the piston, allowed of access to the gudgeon and connecting-rod end, and preserved an equal area of piston exposed to steam both at back and front. The direction of motion of the crank-pin, when the engine was moving "ahead," was so arranged that the thrust of the connecting-rod was upward; consequently, the pressure on the cylinder walls was the difference, instead of the sum, of the weight of the piston, &c., and the thrust of the rod.

This engine was the lightest and most compact of all the forms of marine screw engines, when constructed of the same materials; and for very large sizes with the lower steam pressures has been unsurpassed by any other type of engine. The length of stroke was considerably more than that of the ordinary direct-acting engine, and the connecting-rod much longer than that of any other form, being from two and a half to three times the length of stroke; the weight of the piston was taken by the trunks in great measure, and there were no piston-rod guides. But with the increase of pressure the defects of this form became more apparent, and lie with the very part that distinguishes it—viz., the trunks. The friction of the large stuffing-boxes was very great; in fact, may be so great by unduly tightening the glands as to stop the engine. The loss of heat from the large surface of the trunks being alternately exposed to steam and to the atmosphere was very great, as was also that from their inner surfaces. The gudgeon brasses were exposed to a very high temperature and liable to become heated, and when heated not easily to be cooled, and were not accessible for adjustment; and, perhaps not least, the cylinders became oval from the thrust of the connecting-rod exceeding the weight of the piston when about mid-stroke, and in that position the trunks render least help, and when the engines were going "astern" both the thrust and weight were on the bottom side of the cylinder, and were united in rubbing away the metal.

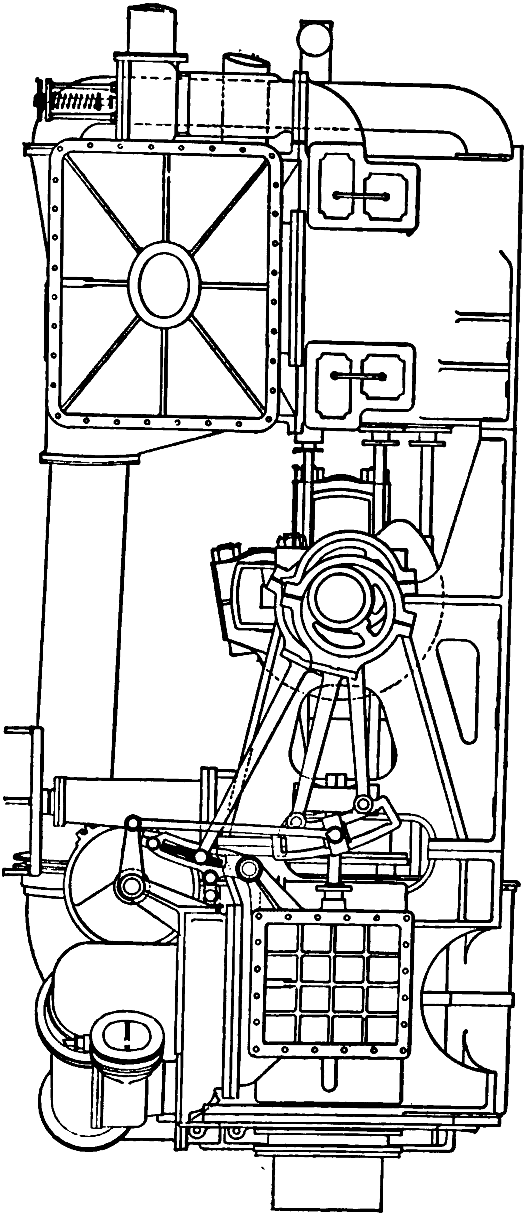

Fig. 14.—Trunk Engines.

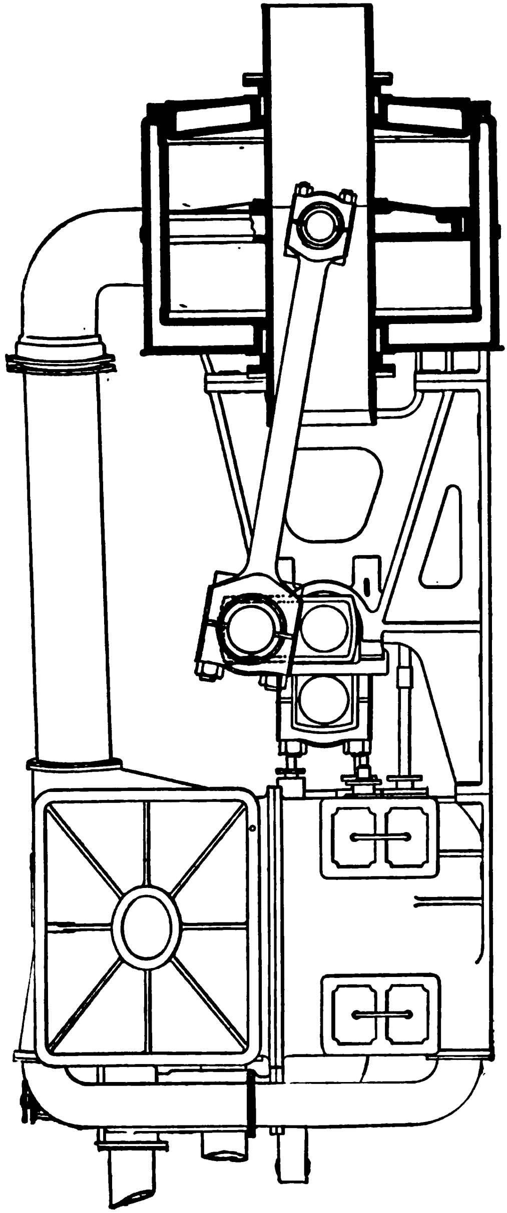

Fig. 14a.—Trunk Engines.

The Return Connecting-Rod Engine (figs. 15 and 15*a*) was the third form of the horizontal type formerly in general use. It differs chiefly from the others by having the connecting-rod on the opposite side of the crank-shaft to the cylinder, there being two piston-rods, one above the shaft on one

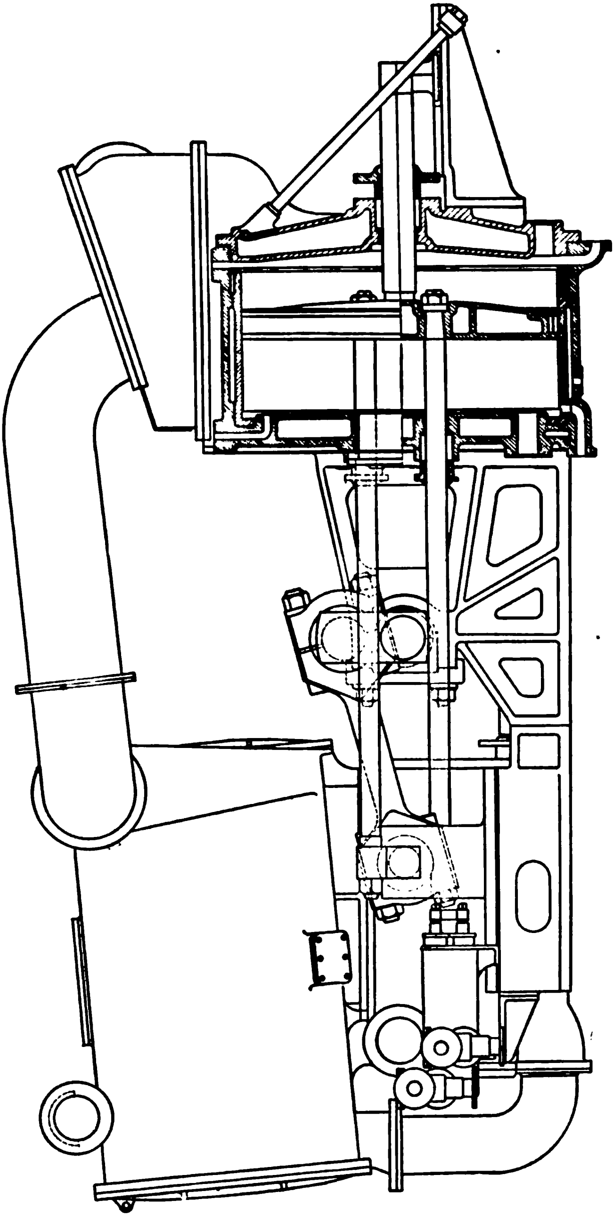

Fig. 15.—Return Connecting-Rod Engine.

side of the crank, and the other below the shaft on the other side of the crank, and both secured to a crosshead, to the centre part of which the connecting-rod is coupled. This arrangement admits of a much longer stroke than either of the other plans, as the cylinder may be so close to the shaft as to allow only for the clearance of the crank, and of a long connect-

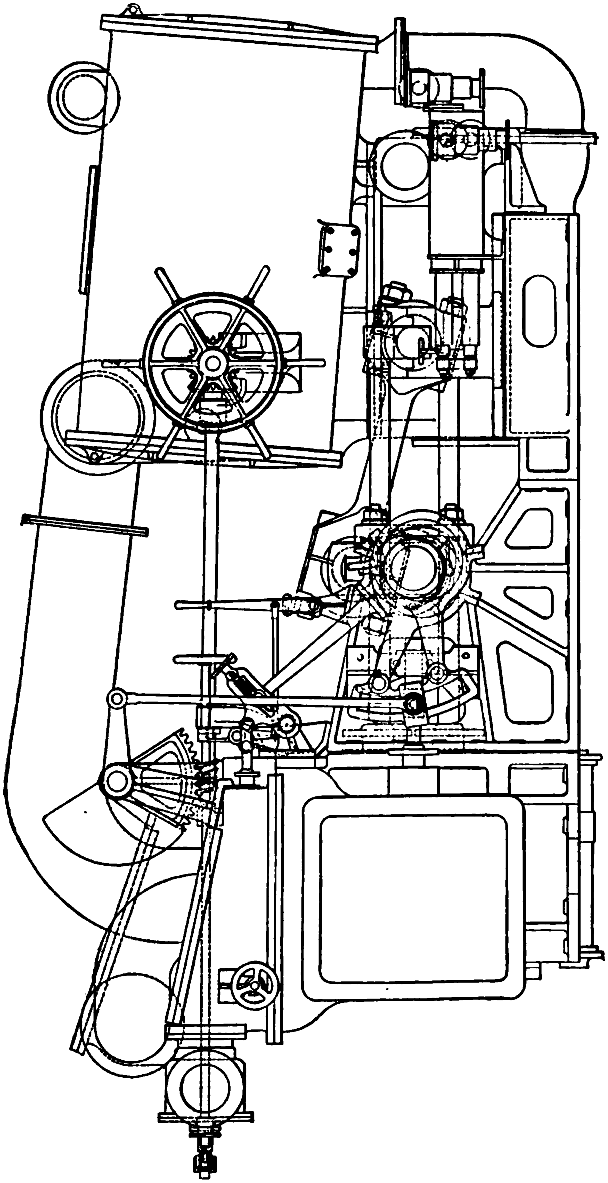

Fig. 15a.—Return Connecting-Rod Engine.

ing-rod, since there need be nothing beyond the crosshead to obstruct its travel. But this is only true, so far as stroke is concerned, in the case of large engines, as, in order to allow for the two rods, there is a minimum limit to the diameter of the piston, and if the capacity of cylinder be limited, then the length of stroke must be made to suit this condition. Hence, in small compound engines it was very difficult to preserve a reasonable ratio between the two cylinders, and so it was found in practice that with these engines the ratio was two and three-quarters to one, and even as low as two and a half to one, thereby very much reducing the good results obtainable from the compound system. One palpable objection to this form is the double rods, necessitating double the number of stuffing-boxes, and doubling the number of parts liable to derangements, besides preventing the main bearings being carried close to the crank-arms, as they should be. The eccentric-rods, also, are necessarily exceedingly short, unless placed, as was done by some makers, on the same side as the connecting-rod, and the working parts generally were somewhat cramped.

It is from considerations such as herein sketched out, that a designer must choose the kind of propeller to suit his requirements, and the type of engine to drive that propeller. None of them are altogether free from defects, and the most successful selection will be that one in which the good qualities are most necessary for the particular service, and the bad ones least harmful, and, as far as possible, mitigated by careful provision in working out the design. The reputation and professional enjoyment of the engineer largely depend on his skill in successfully meeting the deficiencies in the design.

A very large vertical type is shown in Fig. 11, one of the two engines of the screw steamer "Kaiser Wilhelm II.," which has 40,000 H.P. in all, and a speed of 23½ knots. This great power is obtained by increasing the number and size of the cylinders. Fig. 12 represents a modern three-cylinder naval engine largely formed of steel. The large power is obtained by high revolutions and high steam pressure. Fig. 13 shows a modern torpedo-boat destroyer engine, in which a very high power is obtained from light engines by combining an extremely high number of revolutions, a large referred mean pressure, great care in the design of every part, and an unusually low factor of safety.

CHAPTER II.

HORSE-POWER—NOMINAL AND INDICATED.

WHEN the steam-engine began to replace other motors, it was soon found necessary to introduce some unit by which its power could be expressed without using such high numbers as to place it beyond the grasp of ordinary minds. As the engine was frequently taking the place of horses to work mining and other machinery, it was only natural that the work performed by a horse should be taken as the basis for this unit of measurement. The number of units of work performed by the most powerful dray horses was found to be 33,000, so Watt chose this as the unit of power for his engines and called it "horse-power," and this has continued to be the standard ever since, both for land and marine engines.

Watt found that the mean pressure usually obtained in the cylinders of his engines was 7 pounds per square inch. He had also fixed the proper piston speed at 128 × $\sqrt[3]{\text{stroke}}$ per minute, and his engines were arranged to work at this speed, so that he estimated the power which would be developed when at work to be

$$\text{Area of piston} \times 7 \times 128 \times \sqrt[3]{\text{stroke}} \div 33{,}000.$$

The power so calculated was called "*Nominal,*" because the engine was described as of that power, and in practice that power was actually obtained. But when the boiler could be constructed so as to supply steam above the atmospheric pressure, and the engine was run with more strokes per minute than before, the power developed exceeded the nominal power, and from the name of the instrument by which the pressure of steam in the cylinder was obtained it came to be called the "*Indicated*" Power.

The discrepancy between Nominal and Indicated Power has become so great, that for all scientific purposes the former is utterly useless, and is practically obsolete. It remains, nevertheless, in full force among some manufacturers and users of engines, because it better conveys the commercial value and size than does the actual power; for since the area of the piston is usually the only *variable* in the expression, it follows that the size of the cylinder, and therefore the size of all the other parts, must vary directly with the Nominal Horse-Power. But since Indicated Horse-Power depends on three functions—viz., area of piston, speed of piston, and the pressure of steam—the value may be altered by altering the value of one or more of these, which alteration may be material without altering the commercial value. For example, an engine may be made to run at a much higher number of revolutions, even so as to double its Indicated Power, without any additional cost whatever in construction.

The Admiralty modified Watt's rule so as to suit it to the practice of the early days of steam-navigation, by substituting the actual piston speed for the arbitrary one, and so

$$\left.\begin{array}{c}\text{Admiralty Nominal}\\ \text{Horse-Power}\end{array}\right\} = \frac{\text{Area of piston} \times \text{speed of piston} \times 7}{33{,}000}.$$

But for many years past the Admiralty have dropped the use of the expression altogether, and before doing so finally, only used it in the modified sense of being one-sixth of the Indicated Power.

In the Mercantile Marine the present rule for Nominal Horse-Power is by no means uniform, every firm of manufacturers having its own, and all more or less arbitrary. Before the introduction of the compound engine and increased boiler pressures, it was a very general rule to allow 30 *circular* inches per N.H.P.—*i.e.*, the rule was—

$$\text{N.H.P.} = \frac{(\text{Diameter of cylinder in inches})^2}{30},$$

and after that period for compound engines—D being the diameter of the low-pressure cylinder, and d that of the high-pressure—

$$\text{N.H.P.} = \frac{d^2 + D^2}{33}.$$

But neither of these rules took into account the length of the stroke or the boiler pressure. To provide for the latter some manufacturers decreased the divisor from 33 to 30, as the pressure increased from 60 lbs. to 80 lbs. per square inch; while others contented themselves with the compensation obtained by decreasing the size of the high-pressure cylinder, as the pressure was increased. At the present moment the above rules hold good in certain districts, and so long as there is a recognised standard for the length of stroke they are very convenient in many ways. The competition among engine-builders, however, gradually destroyed all traces of such a standard, and the rivalry between the builders of long-stroke and the upholders of short-stroke engines prevented the possibility of its existence. The general adoption of the triple compound engine, however, caused the question of stroke to be removed from the field of competition, and there is now again more uniformity in practice. Besides which, in some large centres of marine engineering all the makers have a standard set of sizes of cylinders for N.H.P. to which they adhere. In order to meet the difficulties some engine-makers have adopted a rule for Nominal Horse-Power, based on the *capacity* of the cylinder, and in so doing have nearly met the requirement on which the continuance of the expression depends—viz., the measure of the commercial value of the engine; but it fails, as compared with the old rule, to maintain a fixed or nearly fixed ratio between the N.H.P. and the I.H.P. For the power *per revolution* depends on the capacity of the cylinder so long as the mean pressure is the same; but since small engines are usually worked at a higher number of revolutions than larger ones, the power developed by the former will bear a larger ratio to the Nominal Power than will be the case in the latter.

A simple and fair rule for N.H.P. is deduced as follows:—D is the diameter of low-pressure cylinder, d that of the H.P., d_1 that of the M.P., and d_2 that of the second intermediate of a quadruple engine; S is the stroke in inches, and D × 0·65 is the standard length of stroke. $\left(\frac{D}{d_1}\right)^2 = 3$; $\left(\frac{D}{d_2}\right)^2 = 7$ or thereabouts in a triple compound; in a quadruple compound $\left(\frac{D}{d}\right)^2 = 8$; $\left(\frac{D}{d_1}\right)^2 = 4$; $\left(\frac{D}{d_2}\right)^2 = 2$ or thereabouts.

$$\text{N.H.P. (Triple)} = \frac{d^2 + d_1^2 + D^2}{30}$$

$$\text{N.H.P. (Quadruple)} = \frac{d^2 + d_1^2 + d_2^2 + D^2}{30}.$$

Any change of stroke should give a proportionate change of N.H.P., hence

$$\text{N.H.P.} = \frac{d^2 + d_1^2 + D^2}{30} \times \frac{S}{D \times 0{\cdot}65}.$$

Substituting the values of d, d_1 and d_2 in terms of D, then

$$\text{N.H.P.} = \frac{D \times S}{13} \text{ for a triple-compound engine.}$$

$$\text{N.H.P.} = \frac{D \times S}{10{\cdot}5} \text{ for a quadruple-compound engine.}$$

Similarly

$$\text{N.H.P.} = \frac{D \times S}{15} \text{ for a compound engine.}$$

No *Nominal* Power can be any guide to the capabilities of the engine, unless the power of the boilers is also in some way expressed or understood; and as it is not easy to imagine how the former can be introduced into any expression which shall effect the latter, or *vice versâ*, the requirements of Lloyd's Committee have remained unfulfilled, and the Register now contains a statement of the leading particulars of the boilers.

That there is need of uniform practice in naming the power of engines, is apparent to everyone having to do with steamships, and the Board of Trade Department, which registers the power, tried to do this by taking N.H.P. $= \frac{d_1^2 + d_2^2 + d_3^2}{30}$,—the old formula. As this takes no cognisance of stroke it was never satisfactory; the Department, however, is still satisfied with it for its purpose, but it is surely time for it to find some other rule which shall determine the rating of engineers and other such matters, as well as be a fair indication of the power the ship has to propel her.

Estimated Horse-Power.—As it is necessary that a power be named for an engine which shall enable unprofessional men to judge of its capabilities, the better plan would be to revert to the principle of Watt, who, as has been shown, attempted to define the power which the engine was actually expected to develop, and have some rule, therefore, which shall give approximately the Indicated Horse-Power. It would, of course, be far better to register the I.H.P., but as it is not always possible to obtain this, the next best method is to estimate it, and call it the Estimated Horse-Power, or E.H.P.

The following rule will give approximately the horse-power developed by a compound-, triple-, or quadruple-expansion engine made in accordance with modern practice:—

$$\text{E.H.P.} = \frac{D^2 \times \sqrt{p} \times R \times S}{7500}.$$

Where D is the diameter of the low-pressure cylinder, p the boiler pressure, R the number of revolutions per minute, S the stroke of piston in feet.

For Example.—To estimate the Indicated Horse-Power of an engine having cylinders 30 ins., 48 ins., and 80 ins. diameter and 48-ins. stroke, revolutions 75, and boiler pressure 180 lbs.

$$\text{E.H.P.} = \frac{80^2 \times \sqrt{180} \times 75 \times 4}{7500}$$
$$= 3430.$$

Engines designed for economic working and having a high rate of expansion do not develop so much I.H.P. as the E.H.P. found by this rule; but the difference will not exceed 10 per cent.; it must also be borne in mind that the engines could indicate the E.H.P. if the cut-off was increased.

As some shipowners still enquire for engines of a certain nominal horse-power, and this is generally based on an allowance of so many *circular* inches per horse-power, the following rules may be followed for finding the diameters of the cylinders. Let D be the diameter of the low-pressure cylinder, and d that of the high-pressure, r the ratio of the capacity of the low to the high, and n the allowance of circular inches:—

$$\text{N.H.P.} = \frac{d^2 + D^2}{n}.$$

Now, since

$$D^2 = r d^2;\ \text{N.H.P.} = (d^2 + r d^2) \div n,$$

and

$$\text{diameter of high-pressure cylinder} = \sqrt{\frac{\text{N.H.P.} \times n}{1 + r}}$$

and

$$\text{diameter of low-pressure cylinder} = d\sqrt{r}.$$

Example.—To find the diameter of the cylinders of a triple compound engine of 200 N.H.P.; the ratio of low-pressure cylinder to the high-pressure being 6, and the allowance 30 circular inches :—

$$\text{Diameter of high-pressure cylinder} = \sqrt{\frac{200 \times 30}{1 + 6}} = 29{\cdot}27 \text{ inches},$$

$$\text{diameter of low-pressure cylinder} = 29{\cdot}27 \times \sqrt{6} = 71{\cdot}7 \text{ inches}.$$

Many other rules have been propounded for N.H.P., some of which are ingenious, but impracticable, while others fail to give results of any value whatever, so that neither class needs notice here; but it may be mentioned that when non-condensing engines were more used in steamships than they are at present, it was found necessary to have a special rule for them, which was

$$\text{N.H.P.} = \frac{D^2 \times \sqrt[3]{S}}{20}$$

D being the diameter of the cylinder in inches, and S the stroke in feet.

Indicated Horse-Power may be defined as the measure of work done in the cylinder of a steam-engine, as shown from the indicator-diagrams, and only falls short of the actual work by such small losses as are caused by the friction of the pin or pencil against the paper, the friction of its working parts, and that in the pipes or passages connecting the indicator to the cylinder. The latter discrepancy is by far the most important, and is sometimes serious in very long stroke engines, where the indicator pipe is several feet long. The others, in the hands of a skilful operator, are not so serious, certainly not in marine engines to the extent stated by Mr. Hirn, who says he found the Indicated Horse-Power, owing to losses in the diagram from the friction of the indicator, to correspond with the *useful work* done by the engine.

The Indicator-Diagram.—The diagram itself shows only the pressure of steam acting on the piston at any and every part of its stroke; but from it may be calculated the mean effective pressure acting during that stroke, and it is assumed that the particular diagram measured is only a sample of what might have been taken at every stroke, so that the mean pressure thus calculated is the force acting on the piston during the whole period of its motion in which the power is taken—usually one minute. Hence, Indicated Horse-Power = area of piston in inches × mean pressure in lbs. per square inch × number feet travelled through by the piston per minute ÷ 33,000.

This, of course, applies only to double-acting engines, as in single-acting engines the pressure is acting only half the time on the piston, and hence, instead of taking the number of feet travelled through by the piston per minute as the multiplier,—*the length of stroke in feet* × *number of strokes per minute* should be substituted.

Mean Pressure.—The mean pressure is usually obtained by dividing the indicator-diagram by a number of equidistant ordinates perpendicular to the atmospheric line, and so placed that the distance of the first and last from the extreme limits of the diagram is half the distance between two consecutive ones; the sum of their lengths, intercepted by the diagram, divided by their number, gives the mean length, and this, referred to the scale on which the diagram was drawn, will give the mean pressure. To illustrate this:—fig. 16 is an indicator-diagram whose length, A X, is, say, 5 inches, and taken with a spring requiring a pressure of 30 lbs. per square inch to compress it 1 inch; so that if M L is 2 inches, it represents a pressure of 60 lbs.; and if B L is 2½ inches, it signifies that, at the point L, the pressure on the piston was 2½ × 30 lbs., or 75 lbs. per square inch above the line A X, which, in this case, shall be *the line of no pressure*, and hence is 75 lbs. *absolute*, or 60 lbs. above the atmospheric pressure. Now, for convenience of division, let there be 10 ordinates enclosing 9 spaces—since there is to be a half space at each end, there will be in all equal to 10 spaces—so that the distance between the ordinates is 5 inches ÷ 10, or half an inch. Measure off A B = ¼ inch; B C, C D, D E, &c., each = ½ inch, and at B, C, D, E, &c., draw perpendicular lines, cutting the diagram at M L, O N, &c., Y Z. Then (M L + O N + &c. . . . . + Y Z) ÷ 10 = x inches, and x × 30 is the mean pressure of the diagram.

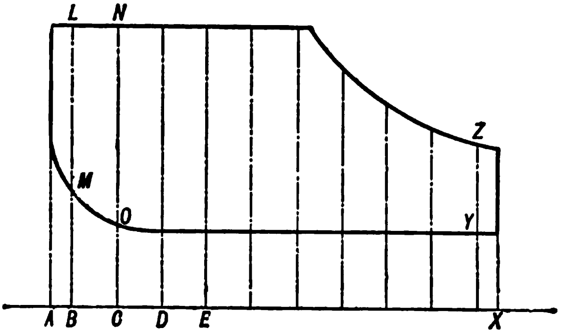

Fig. 16.—Indicator-Diagram.

This diagram is from one side of the piston only, and, when one only is obtainable, it is sometimes assumed to represent both, and the mean pressure thus obtained used to calculate the power; but it seldom happens, although it is much to be desired, that the mean pressure is *precisely* the same on both sides of the piston, consequently, any result obtained in this way is not satisfactory. If the effective area of the piston is the same on both its sides—that is, if there is the same area on which the steam acts to propel the piston forward, on the one side as on the other—the mean pressure found from the diagram taken from the one side, may be added to that found from the diagram taken from the other, and divided by 2 to give the true mean pressure per revolution.

Professor Rankine shows that the diagram should be divided from A to N by ordinates equidistant apart, and the mean obtained by the following rule:—Let n be the number of such divisions (usually 10), $b_0, b_1\ b_2\ b_3\ .\ .\ .\ b_n$ the lengths of the ordinates intercepted by the diagram: then

$$\text{Mean length} = \left(\frac{b_0 + b_n}{2} + b_1 + b_2 + \&c., \ldots + b_{n-1}\right) \div n.$$

The chief objection to this is that, in actual practice b_n would be always without value, and b_0 either without value, or so difficult to measure as to cause differences of opinion as to its value.

Another, and a very ready way of obtaining the mean pressure, is by measuring the *area* of the diagram by a *planimeter*, and dividing it by the length A N, the result being the mean breadth as before, and this multiplied by the scale of lbs. of the spring will give the mean pressure. This is, of course, the quickest plan, and the most accurate, as being mechanical; and where many diagrams have to be calculated with despatch, it is very advisable to have a good planimeter. Special planimeters are now made for this purpose.

In whatever way the mean pressure be measured, it forms the basis of calculation of actual energy, or, as it has been called, Indicated Horse-Power, and is therefore of the utmost importance since most modern formulæ bearing on marine machinery and marine propulsion are based on I.H.P. Hence, any error in taking the diagrams must lead to errors in design from calculations by formulæ based on false premises; this should always be borne in mind by the operator, on whose skill and care a good and true diagram depends as much as on a good instrument. It would be a very valuable quality in an indicator, to be able to give the *useful work* of the engine, as was stated by Mr. Hirn to be the case generally; but it is a quality which no instrument can possess, inasmuch as, with the same *cylinder performance*, there may be a great variety *of actual performances* of the engine, depending on the efficiency of the various parts, and the indicator only gives this cylinder performance. Could the effective power be easily obtained as it is with electric generating engines, a great benefit would be conferred on engineers in making calculations, &c., and in determining the best make and design of engine. The *precise* power absorbed in overcoming the resistance of the working parts of a marine engine cannot be measured; for if the engine be allowed to run without load, it is not running in the *same* state as when running with its load; and any diagrams taken then cannot show what is the loss from friction, &c., on the guides, journals, pistons, and valves when running with the increased pressure on them due to the increased pressure on the piston with the load. Since the efficiency of an engine very much depends on the resistance on these parts, any calculation or formula which excludes, or does not give due allowance to this, is utterly valueless and misleading. Hence, it is false to assume that the power absorbed by an engine in overcoming its resistance is measured by the power indicated when running without load,—as it is possible for an inefficient engine to show a high efficiency when tested in this manner. It is, however, true that a large portion of the resistance is the same, or practically the same, when the engine is running at the same number of revolutions with and without load. The frictional resistance of glands, pistons, pumps, tunnel shaft journals, high-pressure piston valve, and sundry small gear, is the same *per revolution* at any speed, whatever be the load, or power developed. The total resistance or non-useful work of engines, therefore, probably varies with the revolutions.

Efficiency of the Engine.—The efficiency of the marine engine is affected by causes which may be classed under three headings, viz.:—

I Losses as a "heat" engine.
II. Losses as a machine.
III. Losses due partly to mechanical defects and partly to physical causes.

I. **As a Heat Engine**, its efficiency will depend in great measure on the limits of temperature between which it works. The lower limit is fixed from practical considerations at about 160° Fahr., being the temperature due to 5 lbs. pressure *absolute*, which is as low as is convenient for the expansion of steam in a compound engine, but lower generally obtains with turbines. If steam of 80 lbs. gauge pressure, or 95 lbs. absolute, be expanded eight times, the terminal pressure will be about 12 lbs. absolute theoretically, and about 10 lbs. in practice; the temperature corresponding to 10 lbs. absolute is 193° Fahr., and that corresponding to 95 lbs. is 324°. An engine working under these circumstances will be working between the limits of 324° and 193°, absorbing, so to speak, 131° and rejecting 193°, or rejecting more than it absorbs. Such were the conditions on which the great bulk of marine engines were working when at "full speed," and when working most economically they absorbed another 30° of the 193° otherwise rejected. The loss is not so great, however, as appears at first sight, for since the condensed steam is (by means of the surface condenser) returned to the boiler as feed-water, at a temperature of 120°, it is only so far as *temperature* is concerned 73° at full speed and 43° at economical speeds.

But the steam engine differs from the *theoretical* heat engine in the fact that the *gas*, steam, is converted into the *liquid*, water, in the cycle of operations; and to effect that change it must be robbed of its *latent* heat, or that heat which was consumed in overcoming the molecular attraction of the water, and stored up as potential energy in the atoms of the steam. This amounts to 967° Fahr. at the boiling point, under atmospheric pressure, and 979° at 10 lbs. pressure absolute. Hence, when the ordinary marine engine is working with a pressure of 80 lbs. and expanding the steam eight times, it rejects 193° apparent heat and 979° latent heat, or a total of 1172°, and returns to the boiler again 120°, leaving a balance of 1052° lost; so that each pound of steam gives up 1052° × 772 = 812,144 foot-lbs., or 24·6 horse-power.

In such an engine 2 lbs. of coal per hour will be required for each Indicated Horse-Power, and as each pound of coal will produce about 8 lbs. of steam, the quantity of steam per horse-power per hour will be 8 × 2 = 16; and the quantity per minute 16 ÷ 60, or 0·266 lb. So that 1 lb. of steam from the boiler will produce only 1000 ÷ 266 = 3·76 horse-power in the cylinder; and when it leaves the cylinder it has potential energy equal to 24·6 horse-power, which cannot be utilised. The investigations as to the efficiency of the steam-engine as a heat engine have been fully gone into by Professor Rankine in his work on the *Steam Engine*, and should be thoroughly studied there. Although it is not a branch of the subject which need be dwelt on at any great length in a work of this kind, it is necessary to point out how great a loss takes place from this cause, to account for the apparent inefficiency of the engine when gauged by the consumption of fuel. No attempt to effect economy by saving this rejected heat has met with any great or lasting success.

The Steam and Ether Engine, in which ether was used to cool the steam in the condenser, so that the surface condenser for the steam became a tubulous boiler for the ether—which, when vaporised, did work in a second cylinder, and was itself condensed finally in a second surface condenser (cooled by water)—was a success so far as utilising the waste heat is concerned; but it failed in practice, as can be easily imagined. Its success was based on the fact that ether vaporises at a temperature of 95° Fahr.

The most successful attempts, however, have been those of limited scope, such as making use of waste heat to raise the temperature of the feed-water, when a very small saving of heat is effected, but a great benefit obtained in the increased efficiency of the boiler. Seeing how little can be done at the lower limit of temperature, practical minds have concentrated their energy upon improving the efficiency of the engine by raising the higher limit of temperature.

Trevithick was the first engineer to move in this direction, and, notwithstanding that his engines were non-condensing, and his boilers far from being efficient generators, he succeeded in proving, by their superior economy of fuel over the low-pressure engines of Watt, that steam of high pressure could be used advantageously. This is easily seen when the limit of temperatures of the one kind of engine is compared with that of the other.

It has been stated that Watt expected a mean pressure of 7 lbs. per square inch, when using steam at atmospheric pressure, or 15 lbs. absolute. If the mean back pressure were 5 lbs., the cut-off of steam in the cylinder at about half of the stroke, 7 lbs. mean pressure would be obtained, and consequently the terminal pressure would be 7·5 lbs.; so that Watt's engine may be said to have worked between 180° and 212°, or with a range of 32° only. Trevithick, on the other hand, is said to have used steam at 60 lbs. pressure, and if his engines were arranged to cut off at half-stroke, the terminal pressure would be (60 + 15) ÷ 2 or 37·5 lbs., and therefore would work between the limits of 263° and 307°, or with a range of 44°. Since, however, the steam at commencement of exhaust would, in this latter case, be at a pressure of 22 lbs. above that of the atmosphere, it is probable that so shrewd an observer as Trevithick would perceive this to be very wasteful, especially in a county (Cornwall) having no coal mines, and he would, therefore, expand the steam more than twice. If he did so, even to the extent of three times, the terminal pressure would then be 25 lbs. absolute, and the limits of temperature 240° and 307°, thus getting a range of 67°, or more than twice that of Watt. In making this comparison round numbers only have been taken, and no allowance made for the warm feed-water obtained from the hot-wells of Watt's engine, as the data on which to make comparisons are wanting, especially those from the performance of Trevithick's engines. It is known that Trevithick joined with Woolf in introducing the compound engine, which system was probably suggested by the energy still possessed by the steam when exhausting from the engine invented by the former.

Since Trevithick's day the non-condensing engine has disappeared from competition with the condensing engine, both in pumping water from mines and in driving the propellers of steamships; but in the locomotive and other kindred engines it has been developed and remains unrivalled, because beyond competition with engines requiring condensers. Steam of higher pressures than he ever dreamt of is now being used both in condensing and non-condensing engines; and although at present the limit for marine engines of large power in the mercantile marine is about 200 lbs., there are in the Navy marine engines working at 250 lbs. pressure, of all sizes from 30,000 I.H.P. to 1500 I.H.P.

High Pressure: its Advantages and Disadvantages.—Before the introduction of the compound engine for marine purposes, the boiler pressure had been as high as 60 lbs. in large steamers, notably in H.M. service, with the non-condensing engines of 200 N.H.P., fitted in some line-of-battle ships during the Russian war; but when the compound engine secured the confidence of all classes of steamship owners, that pressure was very much exceeded with beneficial results.

Prior to the general use of the triple-compound engine 90 lbs. was a very common pressure, and many ships' boilers were made for a working pressure of 100 lbs., and a few for as high as 110 lbs. The triple-compound engine was for a long time worked with steam of 150 lbs.; then 165 lbs. became the fashionable pressure, to be soon superseded by 175 lbs.; and now 200 lbs. is very general, although the economic gain by going from 150 to 200 lbs. is questioned by many engineers who have made careful observations of all the conditions involved in the change.

The objection to still higher pressures was one of a practical nature, but has now been overcome, so that superheated steam to 600° Fahr. may be used. Steam at a pressure of 250 lbs. absolute has a temperature of 401° Fahr., or nearly that of the melting-point of tin. It will, therefore, affect the condition of some of the metals with which it comes in contact, rendering the surfaces brittle and in a bad condition to withstand rubbing. Moreover, common unguents are vaporised, and the walls of the cylinders become too hot to condense their vapour when exposed to so high a temperature; the heavy mineral oils now used, however, have a boiling point of 700° Fahr.; the difference of expansion of different metals is so considerable, that the utmost care must be exercised in the design and manufacture of the cylinder, &c., to prevent leakages and breakages. The pressure necessitates considerable thickness in all cast-iron work; and, neglecting all considerations of weight or cost, this alone constitutes a source of objection and danger, inasmuch as the sudden exposure of thick masses of this metal to high temperatures on one side only is sure to distort, and very likely to fracture it. The liability to leakage is, of course, greater with the higher pressure, independent of temperature, and the danger to the attendants, from even small explosions, is very much increased. Again, in order to expand steam at this pressure, so as to obtain from it the maximum power, it will be found necessary to use many cylinders; and although, as will be shown later on, the engine with four stages is not without virtue, a larger number will not commend itself to engineers generally for engines of the smallest and largest sizes. In practice, it has been found that an engine can be worked with steam of much less pressure than mentioned, but superheated to the temperature corresponding to the higher pressure, and be more economic and efficient than a similar one working with the higher pressure of steam.

The advocates of very high pressures seem to overlook the fact that a steamship owner has other means of testing the efficiency of an engine besides that of consumption of coal *per I.H.P.* The owner looks rather to the consumption of coal per ton carried, and he may find that the saving effected in the wear and tear account would often pay for a large amount of coal; further, while immunity from break-down and delay is a subject which no engineer can afford to dismiss under any circumstances from his serious consideration, in many services it is a matter of vital importance. To grasp thoroughly the value of such economics as may, under favourable conditions, be effected, take the case of an Atlantic cargo steamer of 3000 I.H.P., doing a voyage in nine days. Suppose the nominal saving to be effected is two-tenths of a lb. per I.H.P., and that twelve voyages per annum are accomplished. The consumption of fuel should consequently be 600 lbs. per hour less, and the coal saved in a twelvemonth should be 1392 tons, which, at 12s. 6d. per ton, amounts to £870. On the other hand, the wear and tear repair and store account, with engines and boilers using steam of 250 lbs. pressure, will be considerably more than with engines of the same power working at 150 lbs. This, together with the interest and depreciation on the extra cost of the machinery, will probably exceed the above sum of money.

By superheating the steam the temperature is raised very considerably beyond that due to the pressure, and in this way the limit of temperature was extended without increasing the pressure. Steam treated in this way is capable of considerable expansion without liquefaction resulting, even in unjacketed cylinders; and, as the extra heat supplied to the steam is what would otherwise have been wasted, considerable economy can be effected in engines using superheated steam.

Superheating is likely to come again into fashion with good results. Metallic packing in the glands, piston-valves, and carefully fitted pistons have paved the way for it, and at all events removed the chief objections to its use. Electrical engineers have found great advantage both in the slow-running and in the quick-running generating engines at central stations of using steam very highly superheated.

II. Efficiency of the Engine as a Machine.—The marine engine suffers loss, in common with all machines, from certain physical causes beyond the *absolute* control of the most skilful designer, and engineers can only aim at mitigating the evil, without entirely overcoming it. The chief cause of loss of energy is, of course, friction (1) of the piston, (2) of the stuffing-boxes, (3) of the guides and slides, (4) of the shaft journals, (5) of the valves and valve-motion. Another source of loss is that from the resistance of the pumps; and, finally, must be mentioned the inertia of the moving parts, which have a reciprocal action, as in the piston and rods. Unless the momentum is balanced, or the energy imparted at one part of the stroke and stored in the heavy moving masses, is given out wholly by the end of the stroke, a serious loss ensues, and the mechanism has to sustain the strain of forces which might be usefully employed.

1. The Friction of the Piston in the vertical engine depends on the pressure of the packing on the sides of the cylinder; so that, if the piston were solid and simply a good fit, there would theoretically be no friction, and in practice none beyond that due to the viscidity of the unguent and to the pressure on the sides of the cylinder from the rolling and general motion of the ship. This is what is required in a piston, and that one is

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most nearly perfect which is capable of moving steam-tight in the cylinder with least pressure of the packing, and so approximates to the condition of a solid one. Resistance due to this cause has been reduced to a minimum in modern engines by care in manufacture and skill in design; the cylinder is now truly and smoothly bored from end to end, the metal, which should be hard and close-grained, soon becomes polished and glazed, and in the best possible condition for smooth working; the packings of the piston are metallic, and the methods of pressing out the packing-ring such as to ensure a uniform and even pressure of small magnitude. The loss from packing the pistons too tightly may become very great, and too much care cannot be exercised in attending to this most important part of the engine.

In horizontal and diagonal engines, the *weight* of the piston pressing on the side of the cylinder causes friction, and as the cylinder wears in consequence more on the bottom than the top, it gets out of shape, thus necessitating more pressure on the packing-ring to maintain steam-tight contact, and thereby increasing the friction. The arrangements necessary for carrying the weight of the piston prevent all the better forms of piston rings and springs from being adopted in horizontal engines; so that the friction of pistons alone renders this engine less efficient than the vertical form. The most important improvement effected of late years, tending to the better working of horizontal engines, has been making the piston solid and of steel; in this way the weight has been very materially reduced, and the strength increased in the same proportion; the *prevention* by this plan is better than the *cures* attempted by guide-rods, &c.

2. **The Friction of Stuffing-Boxes.**—Since the use of the higher pressures of steam this has become a very considerable factor in the consideration of engine efficiency. It is extremely variable, and depends so much on the care and judgment of the attendants, that, however carefully these parts may have been designed and constructed, their good working is entirely beyond the control of the designer. The manufacture of a good and reliable metallic packing at reasonable rates has removed to a large extent this difficulty, so that now the glands of the high-pressure and medium-pressure cylinders are almost invariably fitted with some form of metallic packing; and even those of the low-pressure cylinder are so treated in all important steamers, especially those of large size, although some engineers still prefer good vegetable packing for them. That the resistance may be very considerable is proved by the fact that the old trunk engines could be stopped by tightening the trunk glands, and even in the ordinary piston-rod engine the speed may be seriously retarded by the same process. The glands should never be so tight that the rods are rubbed absolutely dry in passing through them. For the efficient working of the engine it is better that a faint leakage of vapour should pass out with the rod, as that is generally an indication of the packing being only tight enough, and the moist vapour lubricates the packing and keeps it soft when of vegetable composition.

3. **The Friction of the Guides and Slides** is now, perhaps, the least important in the everyday experience of the losses to which an engine is liable; as those parts may be said to design themselves, and they are in such a position as to command the attention of the engineer; they are also, as a rule, easily lubricated. In most classes of engines the piston-rod guide is the chief one for consideration, and since, from the form of the rod-end, it is nearly impossible to give too small a surface to the *shoe*, it is seldom found,

even in badly-designed engines, to give much trouble; but in certain forms of engine this is not always so, and then care has to be taken both in designing and attending to the guides. The maximum pressure on the piston-rod guide of a marine engine is usually from two to three-tenths of the load on the piston, and supposing the ratio of maximum to mean to be 1·50, and the coefficient of friction, under good circumstances, probably not less than 0·05, then

$$\text{Resistance of guide} = 0{\cdot}01 \text{ to } 0{\cdot}015 \times \frac{\text{load on piston}}{1{\cdot}50},$$

which means that from two-thirds to one per cent. of the energy of the piston is of necessity consumed in overcoming the friction of this one guide.

Improvements in the manufacture, and choice of suitable metals for guides, have very sensibly diminished the loss from this cause; besides which, the pressure on the guides is reduced by making the connecting-rod longer in proportion to the stroke. When cast-iron has become, by rubbing, *glazed* on the surface, there is no material better for guides. However, as some engineers will not wait for, or do not trust to, this state of metallic surface, but prefer the rubbing surface to be of a softer nature than the rubbed, it is not unusual to find white metal fitted to the "shoes." That some of the older engines were inefficient from loss at the guides, is proved by the rapid wearing of the shoes; the work necessary to convert so many cubic inches of metal into powder being the measure of avoidable loss at that point.

4. **The Loss from Friction at the Shaft Journals, &c.**, may be also very considerable, as the pressure on the piston is transmitted from the crank-pin to them, in addition to that caused by the weight of the shaft itself and the connections. The same may be said of the crank-pins, which have pressing on them the whole force on the pistons, in addition to the weight of the rods. This friction is very severe, especially in fast-running engines. It was formerly held that *friction was independent of velocity*, so far as movement through a fixed distance is concerned; that is, if a body be moved through 10 feet, the friction is the same if the movement takes place in 1 second or in 10 seconds; but if time be taken into account, the friction of moving the body ten times over the 10 feet in 10 seconds is ten times that of moving it *once* in 10 seconds. Since M. Morin made his experiments further investigations have been most carefully made by Mr. Tower and others, which show that friction does vary with the velocity, probably as the square root of the velocity; hence for modern piston speeds more surface is required than formerly. In a marine screw engine making seventy revolutions per minute, the friction is seventy times that of one revolution; and, consequently, if a paddle engine, having the same size of cylinders, and working with the same pressure of steam, makes only thirty-five revolutions per minute, its friction of journals will be half that of the screw engine per minute. As the first screw engines working without gearing were generally designed by men whose experience had been gained with the slower working paddle engines, it is not astonishing to find that the bearings were not always sufficient for the work on them, and that the speed of the rubbing surfaces prevented the lubrication from being so efficient as had been the case previously, and so aggravated the evil. Again, the old paddle engine and geared screw engine had cylinders of longer stroke compared with their diameters than had the

direct-working screw engine, and as the diameter of the shaft depends on the area of piston and length of stroke combined, while the pressure on the bearings depends only on the area of the piston, the diameter of the shaft might remain the same, although the size of the piston had been very much increased. Now most of the old rules for length of journals took cognisance of the diameter of shaft only, and although the pressure on the journals might have been doubled, there was only the same surface to take it.

For example, a paddle engine of 5 feet stroke might have the same diameter of shaft as a screw engine of 2 feet 6 inches stroke, each having the same cylinder capacity; but the engine with the short stroke would have a piston area twice that of the long stroke, and consequently with the same steam pressure there would be double the strain on the journals, and this with, generally, double the number of revolutions of the shaft.

Friction and Surface.—It was formerly an axiom that friction was independent of surface; but there must always have been limitations to such a statement. It meant that so long as the moving body was supported on a surface sufficiently large to prevent the unguent from being squeezed out, the resistance due to friction was the same fraction of its weight, whatever the area. Practical engineers, however, never had unbounded faith in this axiom. It is now clearly understood that when two surfaces are separated by a stratum of oil, or other unguent, they do not touch one another, and, therefore, each is sliding on the unguent. It is also generally supposed that the particles of unguent assume a globular form in the process, so that they form a kind of ball-bearing surface. Anyway, so long as the unguent is maintained in position, the resistance is small and is constant. It is, however, well known that when the same unguent is used with different metals having apparently equally good surfaces, there is a difference in the *coefficient of friction*, as the fraction is called; and, further, that certain metals will not work at all well on others—*e.g.*, soft steel on bronze. It is evident, then, that either the metals affect the unguent, or they do not really give equally good surfaces. It is known that oil acts chemically on copper, and, therefore, on copper compounds, while it has no action on tin and antimony. This may account for the different behaviour of white metal and bronze. Modern experiments have shown that surface has its influence, even within the safe limits; that the coefficient is higher with very light loads per square inch than heavier ones, so that when an engine is running light, the percentage of loss from friction is higher than when running full load. In the case of a marine engine running slow, the pressure on the guides, &c., will be much less in consequence of reduced load; so that the coefficient of friction will be, from that cause, somewhat higher. On the other hand, as the engine is moving slower and friction varies, as $\sqrt{V}$, the coefficient will be less, and so one may balance the other.

Tower found that at 90° F.,

$$\text{Coefficient of friction} = 20c\frac{\sqrt{V}}{P}.$$

Where V is the rubbing velocity in feet per minute, P the nominal pressure in lbs. per square inch, c a coefficient depending on the lubricant, which, for sperm oil, is ·0014, rape oil ·0015, mineral oil ·0018, and olive oil ·0019, the oil supply being liberal and constant.

For example, the coefficient for a guide on which the pressure is, say, 100 lbs. per square inch, the piston speed 900, and on which mineral oil is used,

$$F = 20 \times \cdot 0018 \frac{\sqrt{900}}{100} = \cdot 0108.$$

If the oil is sparingly supplied, as with a syphon lubricator, the coefficient may be four times this, or ·0432, which is very nearly what Morin stated to be the best result with well lubricated surfaces.

Temperature also affects the friction; the coefficient is reduced when the temperature rises so as to render the unguent fluid and not sticky. On the other hand, further rises of temperature tend to make it too fluid and thin. With some mineral oils, raising the temperature from 60° to 120° caused the friction to be quadrupled.

The aim of the engineer must, therefore, be to prevent metallic surfaces from coming into *actual contact*, for then the friction would be very severe, and soon cause the surfaces to abrade, and even, in the case of cast iron, to strike fire. The lubricant should be introduced at the points of least pressure in order that those of greatest may be well lubricated. Friction at the journals, too, must always be a source of anxiety, though not nearly so much so now as formerly. Crank-shafts are more truly turned, although even now there is room for improvement; the foundations of the engines are more stiffly made, so that there is no springing at the bearings; the bearing surface is *pro rata* larger, and white metal is universally fitted to both crank-pins and main bearings.

5. **The Friction of Valve Motions** and of the valves is very considerable at all times, and is severe when the valves are running dry. Even when the pressure on the valves is partly relieved by frames, &c., on their back, the strain on the rods is sometimes so great as to bend them. With the largely increased piston speed of necessity have come larger valves. This, with increased boiler pressure, drove most makers of marine engines to revive the piston valve for the high-pressure cylinders, and latterly to fit them to the medium-pressure cylinders. On the other hand, a few of the eminent engine builders have stuck to flat valves for all their cylinders, and that, too, with a success that has not attended the efforts of all who held out against the use of piston valves.

Attempts have been made from time to time to use Corliss and other quick-shutting valves instead of slide valves. So far the results have been disappointing, and the increased efficiency due to quick cut-off and small clearance which such valves and gear give to land engines is now not attainable to the same extent with triple-compound engines. But the marine engine may still have its efficiency very much improved by some form of valve which, while giving plenty of egress for the steam to the condenser, very much reduces the clearance now obtaining in low-pressure cylinders.

6. **Loss from the Pumps.**—In all engines, whether condensing or non-condensing, there is a loss of efficiency from the feed-pump when it is worked by the engine. It is true that the work done by it in forcing the water into the boiler is stored energy, and, therefore, not lost, but its want of efficiency is a source of loss. It is now, however, the common practice for the feed-pump to be worked by an independent engine. This is convenient, and has advantages which will be discussed elsewhere, but it must not be overlooked that the cost is greater than by the old method, as the

independent engine consumes at least double the amount of steam per I.H.P. that the main engines do.

To some extent the air-pump may be said to store energy, for it takes water from under a pressure of, say, 2 lbs. per square inch, and places it at the disposal of the feed-pumps under a pressure of 15 lbs. The chief work of the air pump is to withdraw air and other gases from the condenser and deliver them at atmospheric pressure. This, except as a means to an end, is all lost energy, while the friction, &c., of this pump adds to the loss, and so reduces the efficiency of the engine.

Air-pumps, especially in warships, are sometimes worked by an independent engine; the same objections apply to this pump being dealt with in this way as to the feed-pumps, and there is not the corresponding advantage in practice, although there might be if the pump were regulated to run exactly to the requirements of the condenser.

The circulating-pumps render nothing for the large amount of energy put into them, and, viewed in this light only, would seem to be a great impediment to the engine. As a means to a most valuable end it is otherwise appraised; as a necessary adjunct to the surface-condenser it has conduced very much to the economy of the modern marine engine. Besides all the energy lost in the driving of this pump the greatest waste of the steam engine is caused by it, for its water conveys away all the latent heat required to make the steam. It has been suggested that by turning the stream of water towards the stern some propelling effect would be obtained, and some of this loss be prevented. So far no engineer has gone out of his way to effect this small economy.

7. Inertia of Moving Parts.—There should be no losses from this cause if proper arrangements are made, for all the energy stored in them in getting them to their maximum motion should be usefully employed before their coming to rest. In the older engines, as in many modern ones, considerable loss arose from the fact that much of the energy stored in the pistons and other heavy moving parts was not usefully employed, but was wasted in vibrating the engine itself and the ship. This, however, is fully dealt with in Chap. xxiv. and Appendix B. There will, of course, still be small losses due to friction on bearings and guides from the momentum of the moving parts after the engine has been balanced, for the balancing cannot bring all the forces into one plane.

It will be seen by the foregoing that the energy lost in a marine engine is not a definitely fixed quantity, and is dependent neither on design nor on construction alone, but rather on the degree of care exercised by those having the working of it. That much may be saved by good and careful designing and workmanship is evident, but, within certain limits, a good engine may prove less efficient than a really inferior one from lack of proper attention from those in charge, and especially if they are not supplied with good and suitable lubricants.

III. The Losses due partly to Mechanical Defects and partly to Physical Causes are those which cannot be classed as belonging to the engine as a machine, nor to it as a heat engine simply. The most important of these is consequent on the employment in its construction of materials having a high conductivity for heat. The steam pipes are generally of copper, and, when of steel, are comparatively thin, so that there is much loss of heat from their surfaces. Careful covering with material having a low conductivity does much to prevent this waste, but, with the high pressures now used and consequent high temperatures, unless the lagging is very thick and the pipe flanges well covered there is still great loss. The loss, too, is not limited to the mere heat which escapes, for if saturated steam—that is, steam containing as much water as it can—is robbed of any of its heat on its road to the cylinders it deposits some of its water; this water obstructs the free passage of steam through valves and passages, and, till forced through the escape valves or drain cocks, obstructs the pistons themselves. The efficiency of the engine as a heat engine is also materially affected by the presence of water in the cylinders. The losses are very manifest when the speed of the engine is, as often happens, materially checked by water coming with the steam. The chief gain from superheating the steam for the marine engine arose, and will again be found to arise, from the steam being quite dry rather than from the higher temperature at the cylinder.

Liquefaction takes place on expansion in the cylinders in spite of steam jackets and superheating, and a small amount of water to lubricate the internal parts is a good thing, especially as the use of oils is very restricted or wholly done away with nowadays. The cylinders, however, should be as carefully "lagged" as the steam pipes, and the covers and flanges of the high-pressure cylinder should be as carefully covered as any other part.

It is very doubtful if steam, during expansion in the fast-running marine engine, can take any heat from the jackets; that such jackets add to the efficiency of the engine must, however, be admitted when they are properly drained; it is probable, therefore, that the gain is one of efficiency from the fact that the cylinders are freer of water with steam jackets in use.

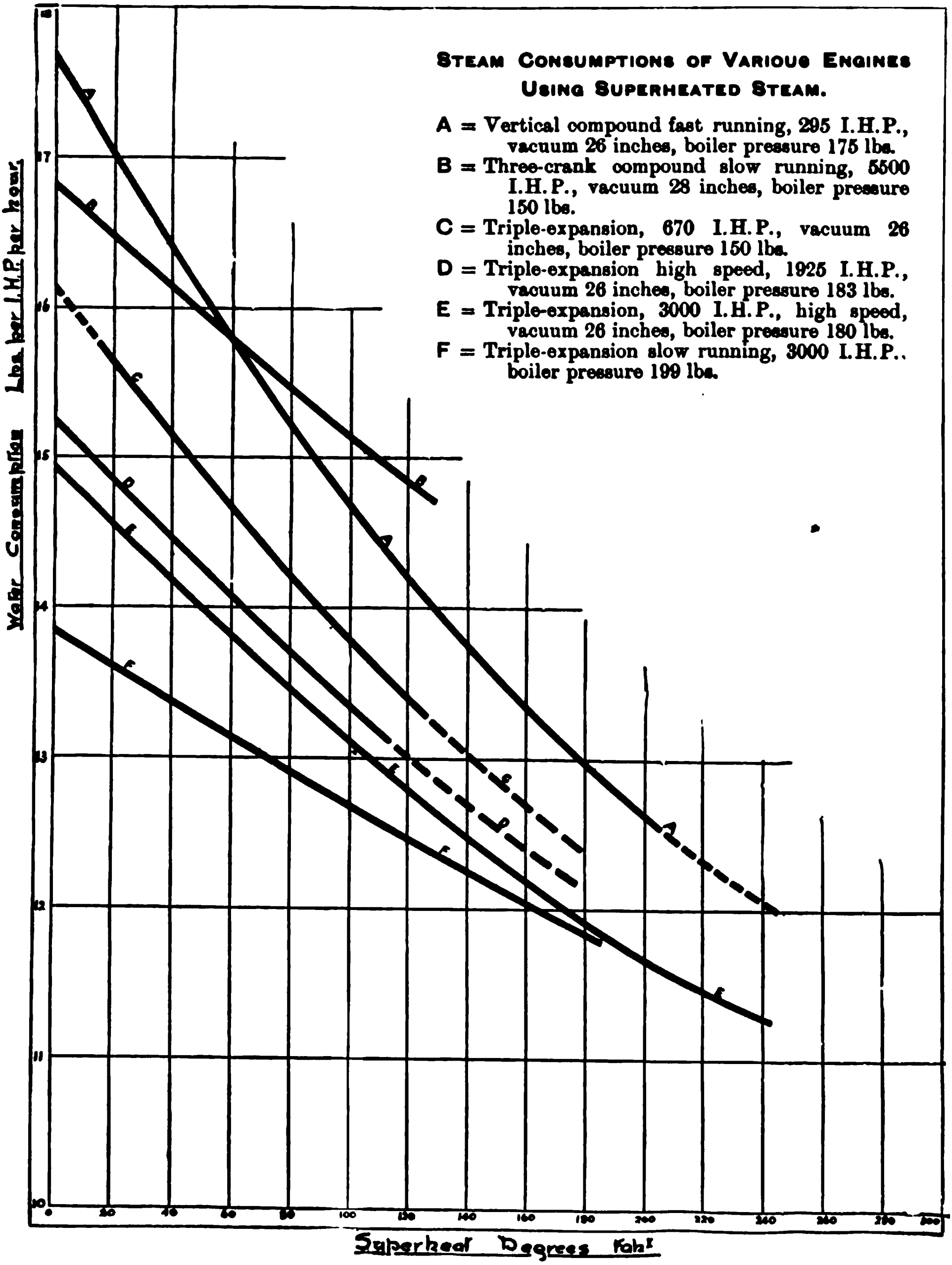
If this argument is based on fact, a jacket to the low-pressure cylinder might be of some use, and outweigh the loss arising from the reheating of steam about to enter the condenser at exhaust.

Superheating the steam so that it might continue wholly in the vapour state till it reaches the condenser has been the hope of many sanguine inventors. Superheating the steam with heat that otherwise is lost is obviously most desirable as a means of economy. Superheating, however, must have its limits, for in economising steam there may be great loss of energy, as, for example, by excessive friction of the high-pressure piston, valves, and rods from want of a lubricant when the steam is so dry as to absorb it all in the vaporous state. As a matter of fact the steam may leave the marine boiler with a large amount of superheat, and even on reaching the cylinder may still have a fair margin, but the piston will not have moved many inches before it is saturated steam. If the steam is to avoid water deposit in the high-pressure cylinder the superheater will require to have its own fire, and unless very special means are adopted for lubricating the piston and valve will work badly and cut the surfaces.

Superheating went out of fashion on the advent of steam of 60 lbs. pressure, partly due to the fact that its temperature, 307° F., was nearly as high as superheating had then gone, and to the restrictions put by the Board of Trade on superheaters.

Metallic packing was unknown, and the gain in economy by the compound engine over the old surface condensing 30-lb. pressure ones satisfied fully the enterprise of the good folks in those days of good freights.

By superheating the steam for electric light installations on shore great reductions in consumption of fuel have been effected (fig. 16*a*). No doubt similar results may also be obtained by employing satisfactory superheating apparatus on board ship, but great care must be taken in the design and



STEAM CONSUMPTIONS OF VARIOUS ENGINES USING SUPERHEATED STEAM.

A = Vertical compound fast running, 295 I.H.P., vacuum 26 inches, boiler pressure 175 lbs.

B = Three-crank compound slow running, 5500 I.H.P., vacuum 28 inches, boiler pressure 150 lbs.

C = Triple-expansion, 670 I.H.P., vacuum 26 inches, boiler pressure 150 lbs.

D = Triple-expansion high speed, 1925 I.H.P., vacuum 26 inches, boiler pressure 183 lbs.

E = Triple-expansion, 3000 I.H.P., high speed, vacuum 26 inches, boiler pressure 180 lbs.

F = Triple-expansion slow running, 3000 I.H.P., boiler pressure 199 lbs.

Fig. 16*a*.

construction of the piston and valves of the first cylinder into which such steam enters, or the loss may exceed the gain in steam efficiency.

Another source of loss is the resistance due to the friction of the steam in passing through the pipes, passages, and valves; and although here again there is not a total loss, still it is not compensated for in the way that the engineer desires. As friction causes heat, so the friction of the steam along the surface of the pipes and passages generates heat; but since this heat is not allowed to escape, it is taken up by the steam, and so superheats it. The loss due to this cause in the steam-pipes is probably very small, especially when the pipes are of such a size that the velocity of steam through them is not excessive, and Mr. D. K. Clark found that it is inappreciable when the velocity is not more than 130 feet per second with very dry steam, and 100 feet per second with ordinary dry steam. The greatest loss is when the steam has to pass through narrow orifices where the perimeter of such orifices is large compared with the area, as is the case at the steam ports of a cylinder; this is called "wire-drawing" the steam, and there is always a loss of pressure from this cause, even when the area through which the steam passes is equal to that of the section of the pipe through which it has previously passed. When the area is reduced, and the perimeter is large, the loss is, of course, still greater, and hence the loss at the ports of a cylinder, where the cut-off is early by means of a common slide-valve, is very considerable, and may amount to as much as 10 per cent. of the pressure, unless the ports are very large, or the travel of the valve exceptionally long. To obviate such an evil, the double and treble ported valves are used, and other plans adopted, whereby increased area of opening to steam may be obtained. If care is taken so as to avoid all unnecessary obstructions to the passage of the steam in the pipes and stop-valves, and there be sufficient opening of the port at the beginning of the stroke, the loss of initial pressure should not exceed 2½ per cent., and in some marine engines there is no appreciable fall of pressure from the boiler to the cylinders. There is also a loss of energy when the steam enters the cylinders from the sudden change in velocity, which will be from 150 feet to 15 feet per second in large engines, and even greater in small engines, where the piston velocity is very much less than 10 feet per second. This cannot be avoided in any way, as it is practically impossible to increase the piston velocity to even one-half that of the steam; and it would be excessively inconvenient to increase the area of ports, &c., so that the velocity of steam should more nearly approach that of the piston. But as the loss from this cause is very slight indeed, no extra cost expended in attempting to avoid it would meet with an adequate return.

A considerable amount of heat is lost by the radiation from those parts which are alternately exposed to the hot steam and to the atmosphere; and this was especially great in trunk engines, where the surface of the trunks is very large, and, being hollow, of course have the inner surface giving off heat as well constantly. Fortunately, the surfaces of the piston-rods, trunks, &c., soon become highly polished, and so do not radiate the heat so quickly as they would were they rough. This loss, too, cannot be avoided, or even reduced to any appreciable extent.

Finally, there is the loss due to the heat conducted from the cylinders, pipes, &c., to the other parts of the engine with which they are connected, and which pass it away by radiation at their surfaces.

CHAPTER III

RESISTANCE OF SHIPS AND INDICATED HORSE-POWER NECESSARY FOR SPEED.

ALTHOUGH, strictly speaking, it is not the province of the engineer to determine the power necessary to drive a ship at a certain speed, but rather that of the naval architect, still it is a point of great importance to the engineer, and one with the investigations of which he should be fully acquainted. Circumstances sometimes require, indeed, that the engineer shall name the power, as the naval architect may submit that, inasmuch as he is unaware of the efficiency of the particular engine to be supplied, he cannot say what *indicated* horse-power will be necessary, but only what *effective* horse-power. Moreover, the subject is one possessing great interest at all times, and sometimes of the utmost importance to the engineer, as the deficiency of speed obtained at the measured mile from that anticipated may be attributed to the inefficiency of the engine and propeller. This charge may be, and often has been, proved to be true; but, on the other hand, it may be without foundation, the blame really belonging to the designer, who has given the ship lines unsuited to the speed.

Value of Trial Trips.—Trial trips are now conducted, both in the mercantile marine and the Royal Navy, with more care and interest than obtained formerly; and it is not sufficient to prove at the measured mile merely that the ship has done the speed expected, or that the engines have developed the power for which they were designed. Both engineers and naval architects anxiously determine whether the speed has been obtained with the minimum of power, and the engineer can satisfy himself on a most important point—viz., the efficiency of the propeller, and, to some extent, the efficiency of the machinery, while the owner, if it be a private ship, is enabled to judge whether he is paying for "big horses" or "little horses." Another point (and one most important to the owner) which, to some extent, is determined on a trial trip is—at what expenditure of fuel a ton of displacement is carried over a mile. It is not an uncommon thing to find that the engine which burns least fuel per I.H.P., does not compare so favourably with others when measured by this latter standard. The apparent contradiction here is not very difficult to understand when fully looked into; it may be, perhaps, best comprehended by taking extreme cases. Suppose the blades of the screw are set so as to have *no* pitch; the engine will work, develop a certain power necessary to overcome its own resistance and that of the screw, but it will not drive the ship an inch; the coal consumption per I.H.P. will probably be somewhat heavier than that of the same engine when working with half its load, but still may be light. Now place the blades fore and aft, so that the pitch is infinity, and although there may be now a large development of power, there will be no appreciable speed—theoretically, none at all. In both these extreme cases the consumption per I.H.P. may be very satisfactory, but the satisfaction would not be felt by the owner. It is manifest, then, that between these two extreme limits of pitch there is some value and one position of blade which will give the best result, so far as economy of fuel for load propelled is concerned. Not only is the pitch of propeller an important function in all calculations relating to the speed of ships, but the diameter has a very important bearing also on the subject, and more than was generally thought previous to the remarkable trials of H.M.S. "Iris."

The Resistance of a Ship passing through water is not easily determined beforehand, as it may vary from more than one cause, and in a way often unanticipated, as has been seen during the trials of the very fast torpedo boats and destroyers. The investigations of the late Dr. Froude on this subject have shown that the older theories were sometimes erroneous, and the old-established formulæ unreliable; and perhaps the best source of information on the intricacies of this somewhat complex subject is to be found in the many able papers read by him, and others since his day, before the Institution of Naval Architects and other learned societies.

When the screw or paddle first commences to revolve, the ship makes no headway, and it is only after some seconds have elapsed that motion is observable. The engine power has, during that period, been employed in overcoming the resistance to motion which all heavy bodies possess, and which is called the *vis inertia*. When the engine is stopped at the end of the voyage, the ship will continue to move, and come gradually to rest, unless otherwise retarded by the reversal of the engine or by check ropes. The ship is then said to have "way on her," a phrase which, in scientific language, means that she possesses stored-up energy, called *momentum*, which is given out, when the engine stops, in overcoming the resistance of the water to the passage of the ship through it. This energy was stored up at starting of overcoming the inertia, and remains stored until there is any retardation in velocity. In this way the weight of the ship helps to preserve a uniformity of motion, as that of a flywheel does to an engine, and therefore it is important that tug-boats should have weight as well as power, to prevent towing in the jerky fashion so often observable. When the *vis inertia* has been overcome, the power of the engine is directed to overcoming the resistance of the water, and wind, if there be any, and in accelerating the velocity of the ship; as the speed increases, the resistance much more increases, until the surplus power available for acceleration becomes *nil*, and the whole engine power is absorbed in overcoming the internal resistances, or those belonging to the engine itself and the propeller, and the external, or that of the ship.

The Resistance of the Water is Twofold.—First, the ship in moving forward has *to displace a certain mass of water of the same weight as itself*, and the water has to fill in the void which would otherwise be left by the ship. The work done here is measurable by the *amount* of water, and since it is equal to the displacement of the ship, displacement becomes a factor in the calculations of resistance. But to effect this displacing and replacing of water with the least amount of energy, it is necessary to do it gently—to set the particles of water gradually in motion at the bow, and let them come gradually to rest at the stern. If it is not done gently, and the water is rudely separated, a wave is formed on either side, showing that energy has been spent in raising the water of this wave above its normal level. Although every ship, however well designed to suit the intended speed, causes these waves of displacement, it is the object of the naval architect to reduce their magnitude as much as possible.

The second cause of resistance to the passage of a ship through the water is the friction between the surface of the ship and the water. Resistance from this cause is generally spoken of as *skin resistance*, and is in well-formed ships much greater than the resistance due to other causes. However fine a ship may be, there is, of necessity, a certain area of skin exposed to the water, and though the displacement be very small indeed, and the section transverse to the direction of motion reduced to a minimum, it is found that a considerable amount of power is required to propel the ship through the water, and that, roughly, the power is proportional to the *wetted surface* at the same speeds. It is from this cause that the older rules for speed, involving only displacement, or area of midship section, together with speed as variables, are found to be so misleading.

Speed Formulæ.—It will be easily understood that the magnitude of the waves of displacement depends both on the form of the ship and the speed at which she is propelled. But, again, the form of the ship depends, to some extent, on the displacement, and it roughly expresses the degree of fineness; so that it may be taken to measure both the *amount* of water displaced and the method of displacing it. Hence it was found that the resistance varied as $\sqrt[3]{(\text{Displacement})^2}$, or generally expressed as $D^{\frac{2}{3}}$. The resistance also varies as the square of the speed, and to complete the expression, which will give a definite value to the resistance of a given ship, it was necessary to multiply the product of the above two variables by a quantity found from practice; and if the law were absolutely correct, this quantity should have a fixed value, whatever the size and form of the ship, and would be a "*constant*" multiplier for all cases. If D_1 be the displacement in pounds, S_1 the speed in feet per minute, R the resistance in foot-pounds per minute, A the constant, then

$$R = D_1^{\frac{2}{3}} \times S_1^2 \times A.$$

Multiply both sides of this equation by S_1, then

$$R \times S_1 = D_1^{\frac{2}{3}} \times S_1^3 \times A.$$

Now $R \times S_1$ is the work done in overcoming the resistance R, through a distance S_1, and is, therefore, the power required to propel D_1 at a speed S_1, and if B is the efficiency of the machinery and propeller combined, so that B × I.H.P. is the effective horse-power employed in propelling, then

$$33{,}000(B \times \text{I.H.P.}) = D_1^{\frac{2}{3}} \times S_1^3 \times A$$

$$\text{I.H.P.} = (D_1^{\frac{2}{3}} \times S_1^3) \times \frac{A}{33{,}000\ B}.$$

Now, it is more convenient to express the displacement in tons and the speed in knots; so that if D and S be substituted for D_1 and S_1, D being equal to $D_1 \div 2240$, and $S = (S_1 \times 60) \div 6080 = S_1 \div 101{\cdot}33$, it involves the introduction of other constant quantities, which do not, therefore, alter the expression, so that the whole of these constants may be replaced by a single constant, C, which will express them. Therefore

$$\text{I.H.P.} = \frac{D^{\frac{2}{3}} \times S^3}{C}.$$

D being the displacement in tons; S, the speed in knots; and C, the so-called constant.

It was also supposed that the resistance would bear a direct relation to the area of section transverse to the direction of motion, as this would be the measure of the channel swept out by a ship: hence the following rule:—

$$\text{I.H.P.} = \frac{\text{area of immersed midship section} \times S^3}{K}$$

K being also a so-called constant.

The above two rules were, for many years, the only ones used by shipbuilders in determining the necessary power for a given speed. Their partial accuracy depended on the fact that the wetted skin varies very nearly with the displacement in ships of somewhat similar form,* and that the proportions of steamships were such that the wetted skin varied nearly with the area of immersed section. Their usefulness depended on the information in the hands of the user, and on his discretion in choosing values for C and K. These rules are still used by many naval architects, and are not altogether set aside by any, as in experienced hands they form a good check on the newer methods, and can be used by themselves with fewer data than are required when rules based on wetted skin are employed. Actual values for C and K are given in Tables iii., iv., v., and vi., on pages 51-55, deduced from the performances of ships on trial trips made with every care; but in choosing values discretion must be exercised that the ship for which a calculation is to be made is somewhat similar in form, size, and speed, to the one whose constants are selected.

Coefficient of Fineness.—To determine the form of a ship, as to whether it is "fine," "fairly fine," or "bluff," it is usual to compare the displacement in cubic feet with the capacity of a box of the same length and breadth, and of depth equal to the draught of water; the coefficient by which the capacity of such a box must be multiplied to give the displacement being called the *coefficient of fineness.* Thus

$$\text{Coefficient of fineness} = \frac{D \times 35}{L \times B \times W}$$

D being the displacement in tons of 35 cubic feet of sea-water to the ton; L, the length between perpendiculars in feet; B, the extreme breadth of beam in feet; and W, the mean draught of water in feet, less the depth of the keel. Strictly speaking, the length should be measured from the stem to aft part of body-post on the water-line, instead of to aft part of rudder-post; but as this dimension is not easy to ascertain without referring to the plans, and the calculation is made for the sake of comparison, rather than as an accurate computation, no inconvenience will arise from this, so long as all the ships under comparison are measured in the same way.

It will be easily seen that the above coefficient only expresses a relation between the cubic contents of the immersed portion of the ship and a box of the same dimension, and gives no certain clue to the fineness of the *water-lines*, which is really what is wanted for consideration in dealing with the question of power for speed.

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* *Note.*—Let L be the length of edge of a cube just immersed, whose displacement is D and wetted surface W. Then

$$D = L^3 \text{ or } L = \sqrt[3]{D},$$

and

$$W = 5 \times L^2 = 5 \times (\sqrt[3]{D})^2.$$

That is, W varies as D .

Two ships may have the same dimensions and the same displacement, and, consequently, the same coefficient of fineness, and yet one may have bluff lines and the other fine—the difference arising from the latter having a flat floor, and the former a high rise of floor. To take an extreme case, the fine ship might have a rectangular midship section, and the bluff one a triangular one; and if the "coefficient of fineness" was 0·5, the bluff ship would have rectangular water-lines, while those of the fine ship would be two triangles base to base.

Now, if a coefficient be obtained by comparing the displacement with the volume of a prism, whose base is the midship section, and height the length of the ship, it will indicate the general fineness of water-lines, and form a guide in the choice of the constants for speed calculations.

$$\text{Coefficient of water-lines} = \frac{D \times 35}{\text{area of immersed mid-section} \times L.}$$

To determine the horse-power necessary to drive a ship at a required speed, it is first of all necessary that she shall have a form suitable for that speed. Now, the form of a ship for a certain speed depends in no small measure on her length. The criterion of form, as already shown, is best obtained by expressing the relation of the displacement of the ship to that of a prismatic body whose section is the same as the midship section of the ship and of the same length. The fraction thus obtained indicates the fineness or otherwise of the waterlines—that is to say, the larger the fraction the fuller the water-line of the ship must be.

It is also true that, for a certain speed, the longer the ship the fuller her lines may be. For example, a ship, for a speed of 20 knots, may have a very much fuller coefficient of fineness if 600 feet long than one of 400 feet for that speed.

Let F be the coefficient of fineness of water-lines, L the length in feet, and S the speed in knots, then

$$F = 0{\cdot}39 \frac{\sqrt[4]{L}}{\sqrt[3]{S}}.$$

If it be desired to determine the highest economic speed for a ship of a certain length and form, it may be found by the following formula:—

$$S = \left(\frac{0{\cdot}39 \sqrt[4]{L}}{F}\right)^3.$$

If it is required to know the shortest length of ship whose form is known for obtaining a certain speed economically, it may be found by the following formula:—

$$L = \left(\frac{\sqrt[3]{S} \times F}{0{\cdot}39}\right)^4.$$

Table I. gives the prism coefficient suitable to the length of a particular ship for a given speed; that is to say, the actual displacement of the ship of a given length and for a certain speed, should not be greater than that given by multiplying the displacement of a prism of the same section as the midship section of the ship and the same length by the factor given.

For example,—A ship 300 feet long, for a speed of 15 knots, should have a displacement not greater than ·658 of the prism displacement—that is to say, she must have a coefficient of fineness of waterlines of 0·658.

TABLE I.—Prism Coefficients of Ships suitable to Length and Speed.

Length.	Speed in Knots.																	
	10·0	10·5	11·0	11·5	12·0	12·5	13·0	13·5	14·0	14·5	15·0	16·0	17·0	18·0	19·0	20·0	21·0	22·0
100 feet, - - -	·575	·564	·554	·546	·539	·532	·526	·519	·512	·506	·500	·489	·479	·471	...	...	...	...
125 ,, - - -	·608	·596	·586	·578	·570	·562	·555	·548	·541	·535	·529	·518	·507	·497	·489	·480	·472	...
150 ,, - - -	·635	·624	·614	·605	·596	·588	·581	·573	·566	·559	·553	·542	·531	·521	·512	·503	·495	·487
175 ,, - - -	·660	·648	·638	·629	·620	·611	·603	·595	·588	·581	·575	·563	·552	·541	·532	·523	·515	·507
200 ,, - - -	·681	·670	·661	·651	·641	632	·624	·616	·609	·601	·594	·582	·571	·560	·550	·540	·531	·522
225 ,, - - -	·700	·689	·679	·670	·660	·650	·642	·634	·627	·620	·613	·600	·587	·576	·566	·557	·548	·540
250 ,, - - -	·720	·707	·697	·685	·676	·668	·659	·651	·643	·636	·625	·615	·603	·592	·581	·571	·561	·552
275 ,, - - -	·737	·724	·713	·703	·693	·683	·675	·666	·658	·650	·643	·630	·617	·605	·595	·584	·574	·564
300 ,, - - -	·753	·741	·730	·719	·709	·699	·690	·681	·673	·665	·658	·644	·631	·619	·608	·598	·588	·578
325 ,, - - -	·765	·754	·744	·733	·723	·713	·703	·694	·686	·679	·672	·656	·644	·632	·620	·610	·600	·590
350 ,, - - -	·782	·771	·760	·750	·740	·730	·721	·712	·703	·695	·687	·673	·659	·647	·635	·623	·612	·601
375 ,, - - -	·795	·783	·772	·761	·750	·740	·730	·721	·712	·704	·696	·681	·667	·655	·643	·632	·621	·610
400 ,, - - -	·809	·796	·784	·773	·762	·752	·742	·733	·724	·715	·707	·692	·679	·665	·653	·642	·631	·620
425 ,, - - -	·820	·808	·796	·784	·773	·763	·753	·744	·735	·726	·718	·703	·689	·676	·663	·652	·641	·630
450 ,, - - -	·832	·820	·808	·796	·785	·774	·764	·754	·745	·736	·728	·713	·699	·685	·673	·662	·651	·640
475 ,, - - -	·843	·831	·819	·807	·796	·785	·774	·764	·755	·746	·738	·723	·708	·695	·682	·671	·660	·649
500 ,, - - -	·856	·842	·829	·817	·806	·795	·784	·774	·765	·756	·748	·732	·717	·704	·691	·680	·670	·660
525 ,, - - -	·867	·853	·839	·827	·816	·805	·794	·784	·775	·766	·757	·741	·726	·713	·700	·688	·677	·669
550 ,, - - -	·877	·863	·849	·837	·825	·814	·803	·794	·785	·775	·766	·749	·735	·721	·708	·696	·685	·676
575 ,, - - -	·887	·873	·860	·847	·834	·823	·812	·802	·792	·783	·774	·758	·743	·729	·716	·704	·693	·683
600 ,, - - -	·896	·882	·869	·856	·843	·832	·821	·811	·801	·791	·782	·766	·751	·737	·724	·711	·700	·690
625 ,, - - -	·905	·891	·878	·865	·852	·841	·830	·820	·810	·800	·790	·774	·759	·745	·732	·718	·707	·697
650 ,, - - -	·914	·900	·887	·874	·861	·850	·839	·829	·818	·808	·798	·782	·767	·752	·739	·725	·714	·703

TABLE II.—Multipliers of $D^{\frac{2}{3}}$ to determine I.H.P.

(When Form of Ship is in accord with Table I.)

Displacement.	Speed in Knots.												
	10	11	12	13	14	15	16	17	18	19	20	21	22
500 tons, - - - -	4·43	6·10	8·05	10·30	12·85	15·80	19·10	22·95	26·55	30·80	36·00	...	...
1000 ,, - - - -	4·20	5·80	7·70	9·90	12·40	15·25	18·55	22·30	25·80	29·86	34·58	40·41	...
1500 ,, - - - -	4·00	5·55	7·45	9·55	12·05	14·87	18·13	21·70	25·30	29·30	33·95	39·40	...
2000 ,, - - - -	3·84	5·38	7·30	9·30	11·80	14·60	17·75	21·23	24·90	28·90	33·50	38·70	45·30
2500 ,, - - - -	3·73	5·22	7·10	9·10	11·50	14·34	17·40	20·80	24·50	28·40	33·10	38·05	44·60
3000 ,, - - - -	3·63	5·11	6·95	8·90	11·25	14·06	17·10	20·45	24·15	28·00	32·70	37·45	44·00
3500 ,, - - - -	3·54	5·00	6·80	8·72	11·05	13·83	16·80	20·15	23·82	27·65	32·30	37·00	43·50
4000 ,, - - - -	3·47	4·92	6·67	8·56	10·85	13·65	16·50	19·87	23·50	27·30	31·90	36·55	43·00
4500 ,, - - - -	3·41	4·85	6·55	8·45	10·70	13·48	16·24	19·60	23·20	27·00	31·53	36·15	42·50
5000 ,, - - - -	3·36	4·80	6·50	8·34	10·55	13·33	16·00	19·35	22·95	26·75	31·20	35·80	42·00
5500 ,, - - - -	3·32	4·75	6·40	8·25	10·42	13·20	15·78	19·10	22·65	26·50	30·90	35·45	41·50
6000 ,, - - - -	3·29	4·70	6·30	8·16	10·30	13·07	15·56	18·85	22·40	26·25	30·60	35·10	41·10
6500 ,, - - - -	3·27	4·65	6·23	8·06	10·18	12·90	15·35	18·60	22·15	26·00	30·30	34·80	40·70
7000 ,, - - - -	3·26	4·60	6·17	7·97	10·08	12·70	15·23	18·40	21·90	25·75	30·00	34·50	40·30
7500 ,, - - - -	3·25	4·56	6·11	7·90	10·00	12·53	15·11	18·20	21·65	25·50	29·70	34·20	39·90
8000 ,, - - - -	3·24	4·53	6·05	7·83	9·92	12·40	14·98	18·00	21·40	25·25	29·40	33·90	39·53
8500 ,, - - - -	3·23	4·51	6·02	7·76	9·84	12·26	14·85	17·83	21·15	25·00	29·10	33·60	39·16
9000 ,, - - - -	3·22	4·49	6·00	7·70	9·76	12·12	14·70	17·66	20·95	24·75	28·80	33·30	38·75
9500 ,, - - - -	3·21	4·47	5·97	7·65	9·70	11·98	14·55	17·50	20·78	24·50	28·53	33·00	38·36
10,000 ,, - - - -	3·20	4·45	5·95	7·60	9·65	11·85	14·45	17·35	20·60	24·30	28·30	32·70	38·00
11,000 ,, - - - -	3·18	4·40	5·91	7·50	9·55	11·70	14·26	17·00	20·30	23·90	27·83	32·15	37·30
12,000 ,, - - - -	3·16	4·36	5·88	7·40	9·45	11·60	14·10	16·75	20·03	23·55	27·45	31·75	36·75
13,000 ,, - - - -	3·14	4·32	5·84	7·33	9·35	11·50	13·95	16·62	19·83	23·25	27·21	31·45	36·45
14,000 ,, - - - -	3·12	4·28	5·80	7·26	9·26	11·40	13·86	16·50	19·65	23·00	27·00	31·20	36·21
15,000 ,, - - - -	3·10	4·25	5·76	7·20	9·17	11·30	13·74	16·40	19·49	22·80	26·80	31·00	36·00
16,000 ,, - - - -	3·09	4·22	5·72	7·14	9·08	11·21	13·63	16·30	19·36	22·63	26·60	30·80	35·80
17.000 ,, - - - -	3·08	4·19	5·69	7·10	9·00	11·13	13·55	16·22	19·26	22·50	26·43	30·60	35·60
18,000 ,, - - - -	3·07	4·15	5·65	7·05	8·93	11·06	13·45	16·14	19·17	22·40	26·30	30·40	35·40
19.000 ,, - - - -	3·06	4·12	5·61	7·00	8·86	11·00	13·37	16·07	19·08	22·30	26·14	30·22	35·20
20,000 ,, - - - -	3·05	4·09	5·58	6·96	8·80	10·94	13·30	16·00	19·00	22·23	26·00	30·05	35·00
25,000 ,, - - - -	3·02	4·00	5·46	6·83	8·62	10·75	13·09	15·78	18·75	21·93	25·58	29·54	34·40
30,000 ,, - - - -	3·00	3·94	5·37	6·73	8·50	10·61	12·93	15·61	18·55	21·68	25·28	29·20	33·90

Table II. gives a list of multipliers for determining the indicated horse-power for ships, whose design is in accordance with the conditions of Table I.

For example,—A ship of 5000 tons displacement is required to have a speed of 14 knots,

$$5000^{\frac{2}{3}} = 292{\cdot}4.$$

The factor for a ship of this length to go at a rate of 14 knots is 10·13, then

$$\text{I.H.P.} = 292{\cdot}4 \times 10{\cdot}13 \text{ or } 2962.$$

The skin resistance is in small steamers the chief resistance, was long ago recognised, and many minds had been turned to give the subject serious consideration. It is seen on reference to the performance of very fine steamers running at high-speeds, that—notwithstanding the extreme care taken in the designing of the hull so as to ensure good results—the amount of power required seems excessive, when compared with that of ordinary steamers at moderate speeds. It follows that either the law of resistance varying with the square of the speed, is not true, and the formulæ for comparison at fault; or the ship and machinery have a low efficiency. Modern research and experiment with torpedo boats and destroyers have shown that, at certain high speeds, from some cause, the resistance increases at a lower rate than that of the square; but in the ordinary sea-going and river steamers, the reverse would seem to be shown, as the constants are very low at the full speeds, and higher at the "half-power" speeds, although still low compared with ships designed to steam at the same speed as given at the half-power trials. This apparent anomaly arises in great measure from the fact that the old rules, as given, take no direct account of the wetted skin, and indirectly only so long as ships are of very similar forms. This can be understood by comparing two ships of similar dimensions, the one with fine hollow entrance lines, and the other with fuller and convex lines. The latter ship will not be suitable for very high speeds, but at moderate speeds the lines (though not so fine as those of the fine ship) will not prevent her from attaining *very nearly* the same speed with the same power as the fine ship. The displacement of the fine ship will be less than that of the other, and by comparing their performance by means of the formula (page 38), she will show to disadvantage. But on looking more closely into the matter, it will be found that the wetted skin of one ship is about the same as that of the other, and consequently the slightly higher speed of the fine ship, with the same power as the bluff ship, is appreciated, and is seen to be due to the finer entrance and more suitable water-lines. Again, fast steamers are usually long as a whole, compared with their mid-ship section, and so when their performance is compared with that of shorter ships, it is to the advantage of the latter when judged by the formula (page 39). But since the commercial value of speed must bear some relation to the carrying capacity of a steamer, the old formula based on displacement still finds favour in the eyes of the owners of cargo ships.

Professor Rankine's Method.—The late Professor Rankine suggested a method of calculating the resistance of a ship, which was based on the resistance of the wetted surface, and which also took into account the fineness of the water-lines. It is as follows:—

Rule I.—Given the intended speed of a ship in knots; to find the least length of the *after-body* necessary, in order that the resistance may not increase faster than the square of the speed: take *three-eighths* of the square of the speed in knots for the length in feet. To fulfil the same condition, the *fore-body* should not be shorter than the length of the after-body given by the preceding rule, and may with advantage be one and a half times as long.

Rule II.—To find the greatest speed in knots suited to a given length of after-body in feet, take the square root of two and two-third times that length.

Rule III.—When the speed does not exceed the limit given by Rule II., to find the probable resistance in lbs.: measure the *mean immersed girth* of the ship on her body plan; multiply it by her length on the water-line; then multiply by 1 + 4 (mean square of sines of angles of obliquity of stream lines). The product is called the *augmented surface.* Then multiply the augmented surface in square feet by the square of the speed in knots, and by a constant coefficient; the product will be the probable resistance in lbs.

Coefficient for clean painted iron vessels, - -	0·01
,, ,, coppered vessels, - -	0·009 to 0·008
,, moderately rough iron vessels, -	0·011 and upwards.

Rule III*a*.—For an approximate value of the resistance in well-designed steamers, with clean painted bottoms, multiply the square of the speed in knots by the square of the cube root of the displacement in tons. For different types of steamers the resistance ranges from 0·8 to 1·5 of that given by the preceding calculation.

Rule IV.—To estimate the *net* or *effective horse-power* expended in propelling the vessel, multiply the resistance by the speed in knots, and divide by 326.

Rule IV*a*.—To estimate the *gross* or *indicated horse-power* required, divide the same product by 326, and by the combined *efficiency* of engine and propeller. In ordinary cases that efficiency is from 0·6 to 0·625 (Rankine, *Rules and Tables*). Marine engines to-day have a combined efficiency of 0·65 to 0·75, and some even higher.

Although the method here proposed has been found to give much more accurate and reliable results than those obtained by the older plans, it is open in practice to two very strong objections. First, it is necessary to have an accurate plan of the ship from which to measure the dimensions required; and second, it is difficult in actual practice to measure accurately the angles of obliquity of stream lines, and the calculation requires more time than can generally be devoted to the purpose. Often the horse-power requisite to drive a ship at a certain speed must be calculated at the time the lines are being got out, and it would be too late to wait for a plan of the ship before getting some idea of the power. Again, the size and fineness of a ship cannot be finally decided upon until the weight of machinery is roughly known; and, as this will depend on the power, it is necessary to approximate to it on very rough and ready information, for which rough and ready rules are more suitable than the more refined ones. Hence, the rules based on immersed midship section and displacement can be conveniently used to obtain that approximation, and the power calculated accurately from the augmented surface afterwards.

Dr. Kirk's Analysis.—A method of analysing the forms of ships, and calculating the Indicated Horse-Power, was devised by the late Dr. A. C. Kirk of Glasgow, and met with much favour on all sides. It is very generally used by shipbuilders on the Clyde and elsewhere for comparing the results obtained from steamers with those obtained from others, and likewise to judge of the form and dimensions of a proposed steamer for a certain speed and power.

The general idea proposed by Dr. Kirk is to reduce all ships to so definite and simple a form that they may be easily compared; and the magnitude of certain features of this form shall determine the suitability of the ship for speed, &c. As rectangles and triangles are the simplest forms of figure, and more easily compared than surfaces enclosed by curves, so the form chosen is bounded by triangles and rectangles.

The form consists of a middle-body, which is a rectangular parallelepiped, and the fore-body and after-body prisms having isosceles triangles for bases; in other words, it is a vessel having a rectangular midship section, parallel middle body, and wedge-shaped ends, as shown in fig. 17.

This is called a *block model*, and is such that its length is equal to that of the ship, the depth is equal to the mean draught of water, the capacity equal to the displacement, and its area of section equal to the area of immersed midship section of the ship. The dimensions of the block model may be obtained by the following methods:—

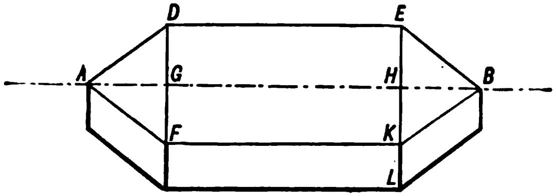

Fig. 17.—Kirk's Analysis.

Since A G is supposed equal to H B, and D F equals E K, the triangle A D F equals the triangle E B K, and they together will equal the rectangle whose base is D F and height A G. Therefore, the area A D E B K F equals E K × A H. The volume of the figure is this area multiplied by the height K L. Then the volume of the block is equal to K L × E K × A H. But K L × E K is equal to the area of mid section, which is by supposition equal to the area of immersed midship section of the ship, and the volume of the block is equal to the volume displaced by the ship. Hence,

$$\text{Displacement} \times 35 = \text{immersed midship section} \times AH;$$

Or,

$$AH = \text{displacement} \times 35 \div \text{immersed midship section.}$$

Now

$$HB = AB - AH, \text{ and } AB = \text{the length of the ship.}$$

Therefore, the length of fore-body of block model is equal to the length of the ship, less the value of A H as found above.

Again, the area of section K L × E K is equal to the area of immersed midship section, and K L is equal to the mean draught of water. Therefore,

$$EK = \text{immersed midship section} \div \text{mean draught of water.}$$

Dr. Kirk also found that the wetted surface of this block model is very nearly equal to that of the ship; and as this area is easily calculated from

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the model, it is a very convenient and simple way of obtaining the wetted skin. In actual practice, the wetted skin of the model is from 2 to 5 per cent. in excess of the ship; for all purposes of comparison and general calculation, it is sufficient to take the surface of the model.

$$\text{The area of bottom of this model} = \text{E K} \times \text{A H}.$$

The area of sides $= 2 \times \text{F K} \times \text{K L} = 2\,(\text{A B} - 2\,\text{H B}) \times \text{K L} = 2$ (Length of ship − 2 length of fore-body) × mean draught of water.

The area of sides of ends $= 4 \times \text{K B} \times \text{K L} = 4\sqrt{\text{H B}^2 + \text{H K}^2} \times \text{K L} = 4\sqrt{\text{Length fore-body}^2 + \text{half breadth of model}^2} \times$ mean draught of water.

The angle of entrance is E B L; E B H is half that angle; and the tangent E B H = E H ÷ H B.

Or, tangent of half the angle of entrance = half the breadth of model ÷ length of fore-body.

From this, by means of a table of natural tangents, the angle of entrance may be obtained.

The block model for ocean-going merchant steamers, whose speed is from 15 knots upwards, has an angle of entrance from 24 to 15 degrees, and a length of fore-body from 0·3 to 0·36 of the length.

For that of ocean-going steamers, whose speed is from 12 to 15 knots, the angle of entrance is from 30 to 24 degrees, and fore-body from 0·26 to 0·3.

Rule.—For angle of entrance of "block model"—

$$\text{Angle in degrees} = 70\,\frac{\sqrt[4]{L}}{S}.$$

L is the length of ship in feet, S is the speed in knots.

Dr. Kirk measured the length from the fore-side of stem to the aft-side of *body-post* on the water-line. This is an unnecessary refinement when screw steamers alone are being compared, as then the length may be taken as that "between perpendiculars." However, when small or moderate size screw steamers are being compared with paddle-wheel steamers, it may be necessary to measure in this way.

Mumford's Method of obtaining approximately the wetted surface of a ship is as follows:—

$$\text{Wetted surface} = L(1{\cdot}7\,D + B \times C).$$

L being the length between perpendiculars; D, the mean draught; B, the extreme beam—all in feet; and C, the coefficient of fineness. It is claimed that by this method a closer approximation to the true wetted surface is obtained than by Dr. Kirk's rule.

To find the Indicated Horse-Power from the Wetted Surface.—In ordinary cases, where steamers are formed to suit the speed as indicated above, the number of horse-power per 100 feet of wetted surface may be found by assuming that the rate for a speed of 10 knots is 5, and that the quantity varies as the cube of the speed. For example,—To find the number of Indicated Horse-Power necessary to drive a ship at a speed of 15 knots, having a wetted skin of block model of 16,200 square feet:

$$\text{The rate per 100 feet} = \left(\frac{15}{10}\right)^3 \times 5 = 16{\cdot}875.$$

$$\text{Then I.H.P. required} = 16{\cdot}875 \times 162 = 2734.$$

When the ship is exceptionally well-proportioned, the bottom quite clean, and the *efficiency of the machinery* high, as low a rate as 4 horse-power per 100 feet of wetted skin of block model may be allowed.

It was observable that ships of H.M. Navy required a large amount of Indicated Horse-Power for their wetted skins, notwithstanding their exceptionally fine lines and smooth bottoms, a fact which would, at first sight, lead to the belief that the large I.H.P. was due to the extreme beam for the length, when compared with ships of the merchant navy. As a rule, the ratio of length to breadth in the merchant service is much larger than in warships; and because the speed constants of the latter are much lower than those of the former, it is urged that long, narrow ships are much easier to drive than broad ones; but this is not of necessity true, and the cause of difference between the two types of ship is not far to seek when carefully looked for.

It is true that the wetted surface of a warship is larger in proportion to her displacement than that of a merchant ship, from the fact that the warship has a greater rise of floor, often a deeper keel, and large bilge-keels or rolling chocks, all of which tend to add to the skin considerably without adding much to the displacement. But although this increase of wetted surface undoubtedly adds to the resistance, and so would account for low constants by the old rules, it is not a sufficient explanation for the high rates per 100 feet of wetted surface.

The friction of the water per square foot of surface may vary slightly with the pressure, so that the resistance from a square foot near the water-line may be different from one 20 feet below it, and as the ships of the Navy formerly *drew more water* for their displacement than do merchant steamers, and the keel and bilge-keels are near the bottom, the average pressure per square foot of their surface will be greater than the latter. But it will be seen that, after all due allowance has been made for this, the performance of the best naval ship was still far behind that of a well-designed merchant ship; and since the ship had had every care bestowed upon the designing, and all allowance had been made for the causes above indicated, it was necessary to seek for the loss in some other direction.

The gross Indicated Horse-Power includes, of course, the power necessary to overcome the friction and other resistance of the engine itself and the shafting, and also the power lost in the propeller. In other words, Indicated Horse-Power does not show by itself, and is, therefore, no measure of the *resistance of the ship*, and can only be relied on as a means of deciding the size of engines for speed, so long as the efficiency of the engine and propeller is known definitely, or so long as similar engines and propellers are employed in ships to be compared. The former is very difficult to obtain, and although a glimpse at the efficiency of machinery may be obtained, as will be shown later on, it is nearly impossible in practice to know how much of the power shown in the cylinders is employed usefully in overcoming the resistance of the ship. Two notable examples from among a great many will illustrate how the efficiency of the *propeller* may vary. H.M.S. "Amazon," originally fitted with a four-bladed Mangin screw, ran 12·064 knots on the measured mile with 1940 Indicated Horse-Power. This result was deemed very unsatisfactory, and immediately after a two-bladed Griffith's screw was fitted, with which the ship made 12·396 knots with only 1663 I.H.P. Many years after this, H.M.S. "Iris" failed to obtain the speed for which she was designed, doing only 16·577 knots with 7503 I.H.P., with a four-bladed screw; but with a two-bladed Griffith's screw 18·587 knots was obtained with 7556 I.H.P., or 2 knots more with the same power. Now, in both these cases four-bladed propellers were replaced by two-bladed Griffith's, and so it may be urged that it was the superior form of blade and the number of blades which made the difference; but in the merchant service four blades is the rule, and the Griffith's form is by no means universal. There was, however, one other point in common with the alterations to the two ships—viz., the pitch was increased when the Griffith's blades were fitted, so that the engines did not make so many revolutions per minute in the case of the "Amazon," and the number of revolutions per knot in the case of the "Iris" was less than before. This is a point very generally overlooked, and the *whole* of the improvement effected, instead of a part only, is thereby placed to the credit of the propeller.*

The engines of a warship were generally horizontal, and of necessity of short stroke, and, in order that a *certain power* might be developed with a fixed weight of machinery, they were run at a higher number of revolutions per minute than is usual in the merchant service; and hence, as pointed out in Chapter ii. it was not surprising to find their efficiency very much lower than the long stroke, vertical, slow running engines of the merchant service.

The success of torpedo boats and destroyers depends almost wholly on their lightness of both hull and machinery, enabling them to do with so small a displacement that they literally skim the water, and the resistance per square foot of wetted skin is consequently comparatively small. Unless small boats are made to float at a very light draught they cannot be driven at high speeds, and all experiments with fast river steamers on the Clyde and elsewhere have shown the decided advantage of light draught. The effect on ships of the depth of water in which they move is also considerable, and very interesting data have been given in the contributions to the Institution of Naval Architects, but it is scarcely a subject for further consideration here.

Progressive Trials.—In modern ship-trials information is usually sought both as to the power required for the highest speeds and (what is equally important) for lower speeds, as such knowledge gives the means of gauging the efficiency of ship and engines, jointly and separately, and is useful in ship designing.

The system of examination is as follows:—Let $P_1 P_2 P_3$ be the power developed in obtaining the speeds $S_1 S_2 S_3$ in knots with $R_1 R_2 R_3$ revolutions per minute. Take a line A N as a base line (fig. 18): on it take points B, C, and D, so that A B, A C, A D are proportional to $S_1 S_2 S_3$; at the points B, C, D erect ordinates, B *b*, C *c*, D *d*, so that they are proportional to $P_1 P_2 P_3$. Through the points *b*, *c*, *d* draw a curve, which is called the *curve of power*, or *curve of I.H.P.*, and it is such that if an ordinate be drawn through any other point, X, on the line A N, the part X *x* intercepted will measure the power corresponding to the speed measured by A X. If the curve is accurately drawn, it will be found that it does not pass through the point A, but above A, at a distance A *a*; this would signify that when the engine was indicating the power measured by A *a* the ship would not move, and so A *a* is the amount of power required to overcome the resistance of the machinery and propeller at starting, or rather, when not propelling the ship, and hence A a is said to represent the *initial friction* of the machinery.

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* Lately the speed of H.M.S. "Drake" has been increased from 23·05 to 24·11 knots, with practically the same I.H.P., by increasing the surface of blades. In the case of the "Iris" the diameter and surface were *both* decreased.

Curve of Revolutions.—A curve of revolutions is constructed in a similar way, by taking points r_1 r_2 r_3 on the ordinates, so that Br_1, Cr_2, Dr_3 are proportional to R_1 R_2 R_3. When the slip is *constant* the curve of revolutions becomes a *straight* line.

Curve of Slip.—The slip may be shown by a curve whose ordinates are proportional to the slip at the speeds S_1 S_2 S_3.

Examination of Curves will show—(1) what indicated horse-power revolutions, and slip correspond to any speed intermediate to those observed; (2) the efficiency of the engine at its *lowest possible speed;* and from it an idea may be formed of its general efficiency, and a comparison made with other engines; (3) the efficiency of the ship, as tested by the rate of increase of power for speed, which is seen by the form of the curve towards the higher speeds—if it begins to mount upwards suddenly it is certain that the resistance has there begun to increase abnormally; (4) that, if the curve is one fairly following the law of resistance increasing as the square of the speed, an estimate may be made from it of the power requisite to drive a similar ship at speeds higher than the highest observed curves, or lower than the lowest; (5) any sudden rise in the slip alone indicates the propeller to be defective in either diameter or surface, or both.

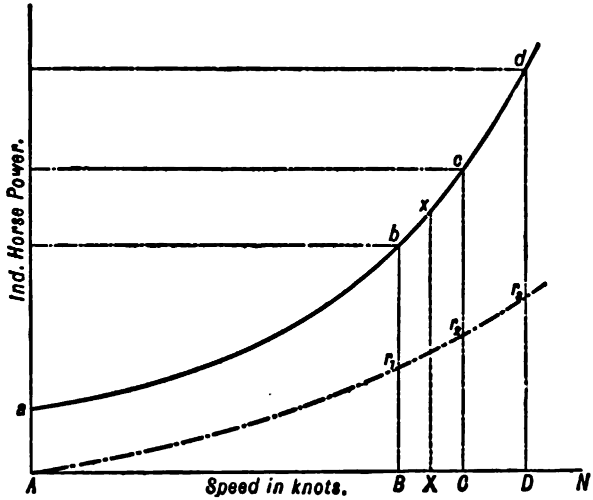

Fig. 18.

Another method of expressing the results of progressive trials is by setting out A B, A C, A D proportional to S_1^3 S_2^3 S_3^3, and erecting ordinates, &c., as before. If the Indicated Horse-Power throughout varies as the cube of the speed, the "power curve" or line drawn through the points b, c, d, will be a straight line; and if the power increases at a higher rate than the cube of the speed at any point, the line will again assume the curved form. The advantages of this plan over the one before described lie in the fact that a straight line is more easily drawn than any curve, that any deviation from a straight line is more easily detected than that of one curve from another, and that the production of a straight line is less liable to error than a curve, so that the interception of *Aa* is less open to error than by the previous method. Of course, the curves of slip and revolutions cannot be examined so well by this latter method as by the former, and it is only the "power curve" that should be analysed in this way.

The values of the different "constants," rates, &c., for ships found from calculations made from the results of carefully conducted trial trips, are more reliable than those got by taking averages when employed in calculations for proposed ships. Tables iii. to vi. give the values of constants, &c., as obtained from the performances of some well-known ships of various types and sizes.

Wetted Skin.—The method of obtaining the Indicated Horse-Power by allowing a certain rate per 100 square feet of wetted surface is by no means a new one, it having been used by some shipbuilders for very many years; but it is comparatively recently that it has been generally used, and now with some reservation, as it requires a considerable amount of accurate information to use with confidence, and a larger staff than was usual in drawing offices, to make the necessary calculations. Although not so accurate as, perhaps, may be considered necessary by some, Dr. Kirk's plan certainly commends itself to all from its simplicity and ease in calculation, and very little experience is sufficient to enable the naval architect to make the requisite allowances for all practical purposes. It will be found that the excess of wetted skin, as calculated from the block model, over that obtained by actual measurement does not exceed 8 per cent., and it is only in exceedingly fine hollow line ships that so great a difference exists. Ships, as generally designed nowadays, having fine lines, but not hollow to the extent that obtained in those constructed on the *wave-line* principle, will, as a rule, differ only by about 5 per cent., and ordinary sea-going steamers by only 3 per cent., while *full* ships will have a wetted surface within 2 per cent. of that obtained by calculation from the block model.

Sea Performance of Steamers.—That the engines may work economically, both in consumption of coal and stores, as well as in wear and tear, it is advisable to run them at such a speed that they develop about 80 to 90 per cent. of the maximum power. Steamship owners do not always care to pay for 25 per cent. more power than is requisite to drive their ships at the speed intended, but there can be little doubt that it is true economy in the end to do so. For short voyages there is not the necessity for this reserve of power, and very fast steamers could not afford to carry the weight entailed by such an excess of power beyond the actual requirements; but ordinary sea-going steamers making long runs can, as a rule, easily do this without much sacrifice, and the wisdom of such a course would be shown by the saving in working expenses at the year's end.

Although, as a rule, trial trips are made honestly, and what the engine has done on the day of trial it can easily be made to do again, still, with the limited staff available for *continuous* service when the ship is at sea, there cannot be that attention devoted to the working parts which was bestowed by the staff of the manufacturer; and the application of water to the bearings and brasses to *prevent* heating, which is usually deemed absolutely necessary by sea-going engineers when the engine is running at full speed, cannot but affect those parts prejudicially, and is a poor substitute for an attendant. Unless, then, the engine is run at a power (*continued p.* 57)

TABLE III.—Results of Trials of Steamships at 9 to 12 Knots.

Designation.	T.S.S. E—. E.	S.S.S. B—. A.	S.S.S. A—. O.	S.S.S. F—. A.	S.S.S. H—. S.	S.S.S. Cv—. D.	S.S.S. T—. E.	S.S.S. Bd—. A.	S.S.S. H—. A.	S.S.S. O—. O.	T.S.S. H—. R.	T.S.S. W—. D.	S.S.S. N—. E.
Length, perpendiculars, ft.,	120·0	200·0	330·0	250·0	255·0	310·0	319·0	320·0	392·0	420·0	460·0	469·0	426·0
Breadth, extreme, - ft.,	21·00	30·00	41·00	32·00	34·50	37·00	34·00	40·00	39·00	48·0	52·00	56·00	54·25
Draught of water, mean, ft.,	7·10	12·10	24·40	12·60	9·20	17·30	17·80	18·33	21·30	18·80	26·70	24·50	24·0
Displacement, - - tons,	282	1285	6880	1740	1420	3600	3445	4720	5767	8160	13,080	13,895	11,556
Midship section, immersed, area,	150	324	925	355	285	590	526	664	738	828	1,322	1,315	1,250
Kirk's Analysis. Wetted skin, - sq. ft.,	4508	8628	25,210	11,184	10,164	18,099	19.245	21,102	26,235	31,196	42,423	43,389	37,798
Kirk's Analysis. Length of fore-body, -	54·00	61·00	70·00	79·0	80·0	96·00	90·00	71·00	118·00	75·00	110·00	99·00	102·5
Kirk's Analysis. Angle of entrance, -	22° 0′	26° 30′	30° 20′	21° 36′	24° 0′	20° 6′	18° 38′	28° 38′	16° 30′	30° 0′	25° 21′	30° 20′	28° 27′
$0·39 \sqrt[4]{L} \div \sqrt[3]{S}$., - - -	·616	·664	·759	·678	·695	·721	·738	·734	·766	·780	·790	·792	·763
$\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$, -	·550	·695	·788	·684	·700	·690	·712	·771	·698	·821	·753	·789	·760
Speed, - - - knots,	9·20	10·8	10·5	11·94	11·30	11·97	11·59	11·79	12·05	11·8	12·05	12·00	12·51
Ind. horse-power, - -	192	661	1372	962	816	1655	1194	1788	1758	2086	3382	3780	3305
I.H.P. per 100 ft. wetted skin, - - -	4·25	7·65	5·44	8·60	8·02	9·14	6·20	8·47	6·70	7·70	7·90	8·71	8·75
Do. reduced to 10 knots,	5·40	6·07	4·70	5·05	5·55	5·32	3·98	5·17	3·83	4·7	4·51	5·04	4·44
$\frac{\text{Displ}^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$, - - - - -	174	228	305	255	223	244	297	258	320	276	276	264	303
$\frac{\text{Immrsd mid sec} \times S^3}{\text{I.H.P.}}$, -	608	617	780	627	504	613	686	610	735	560	690	601	744
I.H.P. ÷ displacement, -	0·680	0·514	0·200	0·553	0·574	0·534	0·347	0·375	0·305	0·255	0·256	0· 272	0·286

TABLE IV.—Results of Trials of Steamships at 12 to 15 Knots.

Designation.	S.S.S. A—. L.	S.S.S. B—. S.	S.S.S. Am—. H.	T.S.S. H—. D.	S.S.S. S—. N.	T.S.S. I—. Y.	S.S.S. L—. H.	T.S.S. A—. B.	S.S.S. C—. O.	S.S.S. B—. L.	S.S.S. B—. B.	T.S.S. D—. B.	S.S.S. A—. O.
Length, perpendiculars, ft.,	300·0	402·0	380·0	550 0	460·0	204·0	240·0	420·0	340·0	400·0	425·00	530·0	300·0
Breadth, extreme, - ft.,	42·00	43·00	43·00	63·00	50·00	34·00	32·00	50·00	41·00	45·20	51·00	59·00	38·00
Draught of water, mean, ft.,	11·20	18·51	18·60	29·92	26·70	10·33	15·40	21·10	17·00	18·60	23·20	23·50	13·10
Displacement, - -tons,	2502	5670	5685	22,589	12,720	1080	2040	9070	4120	5842	9950	14,206	2506
Midship section, immersed, sq. ft.,	401	700	712	1,779	1,270	300	450	980	620	747	1126	1,225	453
Kirk's Analysis. Wetted skin, sq. ft.,	14,623	25,694	24,956	60,090	41,058	7929	12,022	33,065	20,128	26,063	36,448	46,391	14,594
Kirk's Analysis. Length, fore-body, ft.,	81·40	118·50	100·0	106·0	110·00	78·00	81·30	96·00	107·4	126·3	116·0	124·0	107·0
Kirk's Analysis. Angle of entrance, -	24° 52′	18° 0′	21° 32′	31° 36′	24° 25′	20° 30′	20° 0′	27° 16′	19° 36′	18° 2′	23° 42′	22° 47′	18° 36′
$0{\cdot}39\sqrt[4]{L} \div \sqrt[3]{\text{Spd.}}$, - -	·705	·757	·744	·808	·759	·623	·649	·734	·689	·711	·722	·764	·653
$\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$, - -	·729	·708	·737	·809	·761	·617	·661	·771	·684	·684	·727	·765	·643
Speed, - - - knots,	12·23	12·28	12·4	12·75	12·95	13·20	13·23	13·90	14·36	14·79	14·8	14·7	15·33
Indicated horse-power, -	1511	2101	2329	4790	3950	1018	1584	3908	2884	3742	4512	6118	2602
I.H.P. per 100 ft. wetted skin, - - -	10·33	8·17	9·33	8·00	9·46	12·80	13·1	11·79	14·3	14·36	12·35	13·17	17·8
Do. reduced to 10 knots,	5·64	4·41	4·89	3·86	4·36	5·50	5·2	4·40	4·83	4·44	3·81	4·15	4·94
$\frac{\text{Disp}^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$, - - - -	223	279	260	345	299	240	235	306	262	280	299	305	255
$\frac{\text{Immrsd mid sec} \times S^3}{\text{I.H.P.}}$, - -	484	616	581	768	698	685	658	670	633	645	648	637	629
I.H.P. ÷ displacement, -	0·604	0·371	0·410	0·213	0·31	0·942	0·776	0·430	0·700	0·645	0·452	0·430	1·04

TABLE V.—Results of Trials of Steamships at 15 to 18 Knots.

Designation.		T.S.S. C—. R.	T.S.S. Pa—. R.	T.S.S. C—. D.	S.S.S. A—. N.	S.S.S. P—. A.	S.S.S. B—. C.	T.S.S. F—. B.	S.S.S. D—. N.	S.S.S. V—. A.	T.S.S. H—. D.	T.S.S. W—. F.	T.S.S. B—. T.
Length, perpendiculars, - - ft.,		280·0	290·0	300·0	313·5	420·0	450·0	220·0	388·0	466·0	600·0	680·0	570·0
Breadth, extreme, - - - ft.,		31·00	38·00	34·50	33·50	43·00	45·17	28·00	43·00	52·00	65·00	75·00	64·00
Draught of water, mean, - - ft.,		13·40	11·92	13·78	15·00	18·40	23·60	9·00	17·60	20·10	30·01	30·00	27·50
Displacement, - - - -tons,		1940	2100	2200	2480	5906	8500	736	4315	8124	23,630	30,020	19,150
Midship section, immersed area, -		350	416	425	422	698	926	204	615	980	1,810	2,094	1,629
Kirk's Analysis	Wetted skin, - - sq. ft.,	12,582	13,170	14,043	15,200	26,699	32,578	6841	22,330	28,511	63,977	76,440	56,195
	Length of fore-body, - ft.,	86·00	113·3	129·0	107·5	124·0	129·0	94·00	142·0	176·0	143·0	178·0	168·0
	Angle of entrance, - - -	17° 10′	17° 33′	15° 18′	14° 56′	17° 26′	17° 16′	14° 12′	14° 0′	15° 10′	24° 0′	22° 11′	20° 34′
$0·39 \sqrt{L} \div \sqrt[3]{\text{Spd.}}$, - - - -		·646	·653	·644	·660	·714	·728	·590	·676	·711	·766	·787	·737
$\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$, - - .		·693	·609	·603	·658	·705	·714	·573	·632	·622	·762	·738	·723
Speed, - - - - - knots,		15·03	15·00	15·34	15·34	15·10	15·05	16·50	16·82	16·52	16·01	16·20	17·25
Indicated horse-power, - - -		2373	2157	2358	2243	4437	4900	1405	4660	6347	10,508	12,491	11,035
I.H.P. per 100 ft. wetted skin, -		18·6	16·38	16·78	14·75	16·60	15·04	20·40	20·87	22·26	16·41	16·35	19·57
Do. reduced to 10 knots, . -		5·47	4·85	4·66	4·09	4·83	4·42	4·55	4·40	4·94	4·00	3·85	3·81
$\frac{\text{Displ} \times S^3}{\text{I.H.P.}}$, - - - - - .		223	257	259	295	253	289	261	269	287	321	329	332
$\frac{\text{Immrsd mid sec} \times S^3}{\text{I.H.P.}}$, - - -		501	650	650	678	543	642	654	626	696	707	712	760
I.H.P. ÷ displacement, - - -		1·22	1·03	1·07	0·90	0·75	0·58	1·90	1·08	0·78	0·44	0·42	0·57

TABLE VI.—RESULTS OF TRIALS OF STEAMSHIPS AT 18 KNOTS AND UPWARDS.

DESIGNATION.	T.S.S. Pe—. D.	T.S.S. Ch—. D.	T.S.S. Dr—. N.	T.S.S. Ly—. W.	P.S.S. J—. B.	T.S.S. Cl—. K.	T.S.S. Pg—. E.	T.S.S. H.M.S.P.	T.S.S. H.M.S.E.	T.S.S. C—. B.	T.S.S. B—. T.	T.S.S. H.M.S.S.	T.S.S. H.M.S.D.
Length, perpendiculars, ft.,	269·0	300·0	302·0	311·5	245·0	270·0	290·0	300·0	360·0	470·0	685·0	200·0	210·0
Breadth, extreme, - ft.,	33·00	34·50	38·00	36·00	29·00	34·00	38·00	36·50	60·00	58·00	68·00	19·50	20·5
Draught of water, mean, ft.,	8·75	13·78	13·50	11·50	6·10	10·50	11·92	13·90	20·60	20·50	29·91	5·50	5·67
Displacement, - - tons,	1237	2200	2400	1780	690	1350	2100	2235	6,100	9,650	25,910	263	320
Midship section, immersed area,	266	425	460	392	160	311	416	445	1,059	1,100	1,922	77	91
Kirk's Analysis. Wetted skin, sq. ft.,	9678	14,043	14,472	12,628	6976	10,253	13,170	14,083	25,545	35,912	72,208	3890	4323
Kirk's Analysis. Length, fore-body, ft.,	106·3	129·00	119·4	152·5	94·0	118·0	113·3	124·0	158·0	163·0	213·0	80·0	87·0
Kirk's Analysis. Angle of entrance, -	16° 5′	15° 18′	16° 12′	12° 46′	15° 52′	14° 29′	17° 33′	14° 42′	15° 56′	18° 40′	17° 4′	10° 18′	10° 30′
$0{\cdot}39\sqrt[4]{L} \div \sqrt[3]{S.}$, - - -	·593	·617	·613	·617	·576	·589	·590	·590	·616	·670	·725	·486	·475
$\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$, -	·605	·603	·604	·514	·616	·563	·609	·586	·561	·653	·690	·600	·586
Speed, - - - knots,	18·9	18·20	18·57	18·70	19·2	19·3	20·34	20·8	20·96	20·20	20·8	27·60	30·5
Indicated horse-power, -	3231	4398	5241	4000	2760	3750	5820	7275	10,646	16,200	26,500	3628	6017
I.H.P. per 100 ft. wetted skin, - - -	33·4	31·32	36·2	31·67	39·6	36·6	44·2	51·6	41·6	45·1	36·7	93·3	139
Do. reduced to 10 knots,	4·94	5·18	5·65	4·84	5·60	5·09	5·27	5·72	4·52	5·47	4·08	4·44	4·89
$\frac{\text{Disp}^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$, - - - -	246	232	219	240	200	207	236	212	288	230	297	237	220
$\frac{\text{Mid sec} \times S^3}{\text{I.H.P.}}$, - - -	556	583	563	641	411	596	599	551	915	560	652	419	428
I.H.P. ÷ displacement, -	2·612	2·00	2·18	2·25	4·00	2·78	2·77	3·26	1·74	1·68	1·02	13·8	18·8

TABLE VIa.—RELATION OF POWERS AND SPEEDS (SIR W. H. WHITE).

Description of Vessel.	Length in Feet.	Breadth in Feet.	Draught (Mean).		Displacement in Tons.	I.H.P.'s and Admiralty Coefficients.							
						10 Knots.		14 Knots.		18 Knots.		20 Knots.	
						I.H.P.	Coeff.	I.H.P.	Coeff.	I.H.P.	Coeff.	I.H.P.	Coeff.
			Ft.	In.									
Torpedo Boat, - -	135	14	5	1	103	110	200	260	232	870	147	1,130	156
Torpedo Gunboat, "Sharpshooter" Class, - - -	230	27	8	3	785	450	181	1100	203	2,500	190	3,500	186
3rd Class Cruiser, "Medusa," - -	265	41	16	6	2,800	700	284	2100	259	6,400	181	10,000	159
2nd Class Cruiser, "Terpsichore," -	300	43	16	2	3,330	800	279	2400	255	6,000	217	9,000	198
1st Class Cruiser, "Edgar," - -	360	60	23	9	7,390	1000	330	3000	347	7,500	295	11,000	276
1st Class Cruiser, "Blenheim," -	375	65	25	9	9,100	1500	290	4000	298	9,000	282	12,500	278
Atlantic Passenger Steamer, - -	525	63	21	8	11,550	2000	255	4600	304	10,000	297	14,500	281

TABLE VI*b*.—RELATION OF POWERS AND DISPLACEMENTS.*

	No. 1.	No. 2.	No. 3.	No. 4.	No. 5.
Length in feet,	280	300	360	435	500
Breadth in feet,	35	43	60	69	71
Mean draught in feet,	13	16¼	23¾	24½	26¼
Displacement in tons,	1,800	3,400	7,400	11,000	14,200
I.H.P. for 20 knots,	6,000	9,000	11,000	14,000	15,500
I.H.P. per ton of displacement,	3·3	2·65	1·48	1·27	1·09

The following horse-powers were required to drive cruisers Nos. 4 and 5 in the above table at the speeds named:—

	No. 4.	No. 5.
10 knots,	1,500 I.H.P.	1,800 I.H.P.
12 ,,	2,500 ,,	3,100 ,,
14 ,,	4,000 ,,	5,000 ,,
16 ,,	6,000 ,,	7,500 ,,
18 ,,	9,000 ,,	11,000 ,,
20 ,,	14,000 ,,	15,500 ,,
22 ,,	23,000 ,,	23,000 ,,

The frictional resistance of clean painted surfaces varies about as the 1·83 power of the speed, but resistance due to wave making may vary very widely, since it is dependent on form. The total resistance of "Destroyers" has been found to vary as follows: *—

Up to 11 knots, - - - nearly as speed ^{2}
At 16 ,, - - - ,, speed ^{3}
,, 18-20 ,, - - - ,, speed $^{3·3}$
,, 22 ,, - - - ,, speed $^{2·7}$
,, 25 ,, - - - ,, speed ^{2}
,, 25-30 ,, - - practically as speed $^{1·83}$

and the resistances other than frictional vary as follows:—

Up to 11 knots, - - - as speed ^{2}
At 12½ to 13 knots, - - - ,, speed ^{3}
,, 14½ knots, - - - ,, speed ^{4}
,, 18 ,, - - - ,, speed $^{\text{(more than 5th power)}}$
,, 24 ,, - - - ,, speed ^{2}

and at higher speeds as still lower powers of the speeds.

The relation of the frictional to the total resistance is * :—

	"Destroyer."	Cruiser.
At 12 knots,	80 per cent.	90 per cent.
,, 16 ,,	70 ,,	85 ,,
,, 20 ,,	nearly 50 ,,	nearly 80 ,,
,, 23 ,,	...	over 70 ,,
,, 30 ,,	45 ,,	...

If the coefficient of friction be doubled (as it might easily be with a foul bottom), the maximum speed of the "Destroyer" would fall fully 5 knots, and that of the cruiser would be reduced to 19 knots. See Tables iii. to vi*b*.

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* Sir Wm. White, British Association Address, 1899.

somewhat below that of the trial trip, either a larger staff of engineers and attendants must be employed, or the wear and tear may be excessive. It is true, on the other hand, that some engines will develop more power after a voyage or two than was obtained on their trial trip, due to the wearing down of rough surfaces of the guides and cylinder walls, and to the general "smoothing" of all the rubbing surfaces; but it is also true that even such engines should not be run for lengthened periods at their maximum power. Likewise, the improvements in design and the employment of better materials for guides and bearings admit of modern engines being worked at high speeds with less risk than was the case formerly. The better balancing of the engine with three cranks, and the almost perfect balancing of the four-crank engine, together with the extended use of superior white metal in guides and bearings, metallic packing in the principal stuffing-boxes, have permitted of a high rate of revolution with less risk than obtained formerly with the slow-running mercantile engine; but, notwithstanding, a reserve of power for bad weather and emergencies is in every ship highly desirable.

CHAPTER IV.

SPACE OCCUPIED BY AND GENERAL DESCRIPTION OF MODERN MARINE MACHINERY.*

THE choice of the particular type of engine for a particular service depends on many things beyond the control of the designer, and is governed by one or two considerations of paramount importance.

In large ships of moderate power there is free scope generally for designing any particular style of engine which will give good results, or which, in the mind of the engineer, will best suit the conditions of the service; but in small ships, and large ships with large power, the utmost consideration is required in the choice of design, and the highest skill is necessary to work out successfully that design, so as to keep within the limits entailed by the ship's size and construction.

Space occupied by Machinery.—The first condition by which the engineer is bound is the space occupied by the engines and boilers, and, in consequence of this, more ingenuity has been displayed in designing engines to occupy a minimum amount of room than in almost any other direction. The beam of the ship decides the limit of engine-room in one direction, and the requirements of the shipbuilder and owner cramp it in the other. In the merchant navy there is little objection to engines occupying space vertically; but for warships, it is almost a necessity that they be wholly below the water-line or protected by armour. Paddle-wheel engines, which have the shafting in the direction of the beam of the ship, and are thereby limited only to the space occupied by the cylinders and their fittings, are not much affected by this limitation, inasmuch as, although there is sometimes too little room when placed side by side, the cylinders may be situated the one before, and the other abaft the shaft, acting on the same crank, or one above the other, or on separate ones. The stroke of a paddle-wheel engine is only of necessity governed by the dimensions of the ship when the cylinder is placed vertically, and the piston-rod connected directly with the crank-pin, or by means of an intervening connecting-rod. This is the case with the oscillating, and also with the direct-acting vertical engine, the latter being seldom or never used for this purpose. The steeple engine, and other kindred forms, are only partially limited, as even in very shallow steamers there is height sufficient for a considerable length of stroke. The other forms of vertical paddle-wheel engine—side-lever and overhead lever—*may have any length of stroke of piston.*

Long strokes of piston with the oscillating cylinder can always be obtained by placing it inclined in its mid position; and the direct-acting engine having very long stroke, can be employed in the same way. The vertical height of the shaft of a paddle engine above the bottom of the ship depends on the diameter of the wheel, and the draught of water of the ship at the wheel; the diameter of the wheel depends on the speed of the ship and the revolutions at which it is to be run.

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* For particulars as to weight of Machinery, *vide* end of volume.

Breadth of Paddle-Wheel Engines.—The space athwartships of the oscillating engine is generally more than that of any other engine, on account of the room required by the trunnions and their connecting-pipes, as well as by the belts surrounding the cylinders; but notwithstanding this, it is seldom found that an engine of this type cannot be placed even in very narrow river steamers.

The following rules will be found to give approximately the minimum breadth of a paddle-wheel engine, measured over all athwartships *across the cylinders*:—

(*a*) 1 Cylinder oscillating engine - - - - - $B = 2·0 \times D + 1·0$ foot.
(*b*) 2 ,, ,, ,, - - - - $= 4·0 \times D + 2·5$ feet.
(*c*) 1 Cylinder diagonal engine - - - - - $= 1·25 \times D + 1$ foot.
(*d*) 2 ,, ,, ,, (side by side) - - $= 2·5 \times D + 3$ feet.
(*e*) 1 Cylinder steeple engine - - - - - $= 1·25 \times D + 1$ foot.
(*f*) 2 ,, ,, ,, - - - - - $= 2·5 \times D + 2$ feet.
(*g*) 1 Cylinder side-lever engine - - - - $= 1·5 \times D + 1·5$ feet.
(*h*) 2 ,, ,, ,, - - - - - $= 3·0 \times D + 4$ feet.

D is the diameter of the cylinder; when the engines are compound, *each* cylinder is to be multiplied by *half the multiplier* given in the above table.

For example, the breadth of a compound oscillating engine having cylinders d, and D diameter, $B = 2d + 2D + 2·5$ feet.

It often happens, when the engines are of long stroke, that the breadth of the entablature, or frame containing the shaft-bearings, is more than that over the cylinders. This breadth is approximately 6 times the diameter of the crank-shaft for a single-cylinder engine, and 14 times the diameter of the shaft for engines with two cylinders, when there is no intermediate crank to work the pumps; when there is a crank or large eccentric for this purpose, the breadth is about 16·5 times the diameter of the shaft in the journals.

Length of Paddle Engines.—The distance of the cylinder end from the centre of the shaft is, of course, nearly proportional to the stroke, and is in oscillating engines much shorter than in diagonal direct-acting engines. This distance is given approximately, thus:—

(*a*) Oscillating engine with single piston-rod, $L = 1·5 \times S + 0·7 \times D + 1$ foot.
(*b*) Direct-acting ,, ,, ,, $L = 3·5 \times S + 0·6 \times D + 1$ foot.

S is the stroke, D the diameter of the cylinder, which in compound engines is that of the high-pressure cylinder.

Space is saved by fitting two piston-rods, which admits of the crank-pin brasses passing between their two stuffing-boxes; this is sometimes done, but the same economy can be effected in direct-acting engines in a simpler way. The space occupied by lever engines in a fore and aft direction is about three times the stroke of the piston + the diameter of the cylinder. In the other form of engines, where the cylinder is vertical, the space occupied by them depends in great measure on the condenser. The condenser of a diagonal engine is generally placed beneath it, so as to occupy no more space in the ship beyond that covered by the engine itself.

Screw Engines.—The shafting of *screw engines* is in the direction of the length of the ship, and, consequently, the axis of the cylinder or centre line of each engine will be in a plane at right angles to this, and may be vertical, horizontal, or inclined. If the machinery is to be kept under the water-line, it must be horizontal, or only slightly inclined to that position; or, if vertical, of short stroke; while the length from centre of shaft to end of cylinder is limited to the half-breadth of the ship; and, in reality, owing to the rise of floor and round of the bilge, &c., the allowance usually falls far below this. Length of stroke is, therefore, necessarily short by comparison with the vertical engines in the mercantile marine. The vertical engine may have exceptionally long strokes in large ocean steamers; and, even in ordinary merchant steamers, the stroke may be longer than usually obtains, without necessitating any special arrangement in the ship. Roughly, the stroke of a vertical direct-acting engine for a merchant steamer is never less than two-thirds, and may be as long as four-fifths the diameter of the low-pressure cylinder; and that of a vertical Naval engine is 0·5 to 0·6; while that of a horizontal engine, power for power, will be only about a half the diameter of low-pressure cylinder, and even so long a stroke as this was only admissible in ships of great beam, as was usual with armoured ships and cruisers which carry a large spread of canvas.

The Return Connecting-Rod Engine, of all the horizontal kinds, admits of the longest stroke, inasmuch as the cylinder front of it can be closer to the shaft than that of any other type: this distance being limited to the length from the centre of the shaft to the connecting-rod end at the crank-pin; this end of the connecting-rod passes between the two stuffing-boxes of the piston-rods, and hence from the cylinder to it, when at the end of the stroke there need be only sufficient space for safe *clearance*. Space in this class of engine is sometimes saved by *dishing* the piston, which admits of the cylinder fronts being concave towards the rod, and the cylinder itself so much nearer to the shaft. Of course, the cylinder-cover in this case will be convex outwards, and so the distance gained at the front is lost there; but the difficulty is to find space for the cylinder-cover and flange at the turn of the bilge, and by this arrangement such space is obtained, and there is always plenty of room at the level of the centre of the cylinder. The connecting-rod with the piston-rod crossheads, &c., are on the side of the shaft opposite that of the cylinder, and there is never any difficulty experienced in finding room for them. It is usual (and, certainly, necessary) to fit a "tail-rod" or "trunk" at the back of the pistons of horizontal engines having cast-iron pistons, in order to relieve the cylinder bottom of their weight; such rods pass through stuffing-boxes in the covers, and have at their ends slipper guide-blocks supported on slides. There is sometimes a difficulty found in obtaining sufficient room for this apparatus in its most approved form at the aft engine, although generally the form of the sections of a warship admit of it. The general adoption of steel pistons has, however, now obviated the necessity for this.

Trunk Engines.—The distance of the front of the cylinder in these engines is considerably more than that on the preceding type, as there must be sufficient room for the trunk when *full out* to be clear, not only of the crank-shaft, but also when there are balance weights fitted to the crank-arms (as is usual) to be clear of these too. When there are balance weights fitted to the crank-arms, the distance of the cylinder front from the centre of the shaft, is longer by a distance equal to the length of the stroke in a trunk engine, than in a return connecting-rod engine. Again, for the same *area* of piston, the latter has a smaller diameter of piston than the former engine, and, consequently, with the same height of centre the stroke cannot be so long. The back trunk simply passes through a stuffing-box in the cover, and having no guides or appendages, there is generally room for it in the wings.

Length of Screw Engines.—The distance from the centre of crank-shaft to the back of cylinder-cover of horizontal engines is given approximately by the following rules:—

(*a*) Direct-acting engines, - - - - $L = 3{\cdot}5 \times S + 0{\cdot}6 \times D + 6$ inches.
(*b*) Return connecting-rod engines, - $L = 1{\cdot}6 \times S + 0{\cdot}8 \times D + 3$ inches.
(*c*) Trunk engines, - - - - - $L = 2{\cdot}6 \times S + 0{\cdot}6 \times D + 6$ inches.
(*d*) Vertical engines, - - - - - $L = 4{\cdot}25 \times S + 0{\cdot}8 \times D + 6$ inches.

S is the stroke, D the diameter of the cylinder, which, in compound engines is that of the high-pressure; the length of the connecting-rod in the direct-acting engine is assumed to be twice the length of stroke.

The Space required in a Fore and Aft Direction for horizontal engines depends chiefly on the diameter of the cylinders, and the arrangement of the valve-boxes. If these latter are on the sides (that is, if the valve-face is vertical), then, not only is room occupied by the boxes themselves, but space is required to admit of the doors being removed, and the valves withdrawn. This is somewhat reduced by placing the valve-box of the low-pressure cylinder between it and the medium-pressure cylinder, as is usual with vertical engines; but this arrangement is often very inconvenient, and not always admissible.

If the valve-boxes are placed on the top of the cylinders, so that the faces are horizontal, much space is then saved, both in the engine itself, and in the room required for it.

The distance over the cylinders, from valve-box to valve-box, in the former case is

$$3\,D + 1 \text{ foot},$$

and in the latter case

$$2{\cdot}2 \times D + 6 \text{ inches},$$

both being for two-cylinder expansive engines; for compound engines, $1{\cdot}5 \times D + 1{\cdot}5 \times d + 1$ foot, and $1{\cdot}1 \times D + 1{\cdot}1 \times d + 6$ inches, where D is the diameter of the low-pressure cylinder, and *d* that of the high. The distance for three-cylinder engines, with the valves on top of the cylinders, may be found by allowing $1{\cdot}1 \times D + 3$ inches for each cylinder. Space is often saved by arranging the machinery so that the low-pressure cylinder of a compound engine is forward, instead of aft.

Diagonal Engines.—Any engine whose cylinders are not perfectly horizontal may, strictly speaking, be called *Diagonal*. Before vertical engines were generally received as the best type for mercantile purposes, many engineers showed a strong leaning to this particular arrangement; and, until solid crank-shafts could be obtained easily, it was one which commended itself highly for direct-working engines, as two engines at right angles to one another could wosk on the same crank-pin, which could be on a *crank-arm keyed to the shaft*.

The crank so fitted is said to be *overhung*, and is such as is generally used for land engines with success; but as marine engines have cylinders of larger diameter in proportion to the stroke than land engines have, the bending moment on both pin and shaft is very severe, and this was often the cause of break-down of the diagonal screw engine. The remedy was either to fit a solid crank, or else to add another crank-arm and shaft end to the existing one, when the engine worked safely.

Although diagonal engines are no longer made, there were many features in the design which still commend them for consideration in certain special cases. The room occupied in a fore and aft direction is only about one-half of what is required for the ordinary two-cylinder arrangement; the cylinders can, without shortening the length of stroke, be kept under the deck, and the whole of the weight and strain of the engine, when working, is taken directly on bearers attached to the framework of the ship. It is also a cheap and light engine, inasmuch as the crank-shaft and bed-plate are only half the length of those of the ordinary arrangement, and one pair of eccentrics only is sufficient to work the valves. The platforms, ladders, &c., for an engine of this type are also much less, and in many other ways economy is effected.

Some engineers adopted, under special circumstances, a modification of the diagonal engine, in which one cylinder (generally the low-pressure one of compound engines) was placed vertically over the crank, and the other either horizontally or slightly inclined, the condenser being placed on the opposite side; this method was found to give good results, and where space in the ship is to be minimised, it was no doubt a very suitable arrangement.

Engines for twin-screw ships have often to be slightly inclined, to accommodate them to the form of the ship, and allow of the low-pressure cylinder cover being taken off, and the piston withdrawn, without having unduly large engine-room hatchways.

Inverted Direct-Acting Engine.—These engines are now universally adopted throughout the world as the most convenient, and, in every way, the best for both mercantile and warships; and although the varieties of the type differ in details and general arrangement, each manufacturer having his own particular design, yet the same main principle is developed throughout—viz., the cylinders are vertical, with their axes in a plane passing through the keel in single-screw ships, and parallel, or nearly parallel, to the keel in twin-screw ships; each piston-rod end is coupled direct to the crank-pin by a connecting-rod, whose length is now never less than twice the length of the stroke of the piston. The simplest form of this type consists of a single cylinder supported on two columns, having the condenser on one side of the foundation, and the starting gear on the opposite side, the connecting-rod operating on an overhung crank, and a flywheel on the shaft to regulate its motion. Such an engine occupies but little room in the ship, will work very well, and is cheap to construct when of small power.* It is usual for engines of this form to have a solid or two-armed crank to avoid the casualties so common with the older diagonal engines. The condenser, too, is sometimes placed centrally, with the tubes athwartship, so that the centre of gravity of machinery may be in the middle line of the ship.

Engines of this description are called *single-crank single cylinder*.

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* Its suitability for large powers also was tested by Mr. Alfred Holt, who had an engine of this type made with cylinder 40 inches in diameter and 66 inches stroke, worked by steam of 80 lbs. pressure per square inch.

Single-Crank Compound Engine.—By placing one cylinder above the other, and connecting the pistons by a rod or rods, so as to operate conjointly through one connecting-rod on the crank, Mr. Alfred Holt constructed a form of engine which gave very great satisfaction, and by its continued success overcame the prejudice of all who had experience with it. From being the "hobby" of an individual, it was adopted by one or two large steamship companies almost exclusively, and partially by many others. Its principal recommendations are obviously the little space it occupies, and the small number of working parts liable to become deranged or broken. The former is not always an advantage to the steamship owner, owing to the rules for measuring tonnage, as the 32 per cent. reduction could not easily be obtained with a single-crank engine. Nor, on the other hand, is the latter advantage always appreciated by the engineer as it should be, although it offers a guarantee of safety: in case of the breaking of a rod or pin, the whole of the machinery may be rendered useless, but with this form of engine, the number of such parts being reduced, there is a corresponding reduction of the chances of such a casualty happening. The introduction of the triple compound system, however, caused the construction of this class of engine almost to cease. To compare the safety of a single-crank with that of a double-crank engine, it is interesting to look closely into the matter.

The condenser, pumps, shafting, and propeller being common to the two engines, they may be dismissed from consideration. In the single-crank engine there is, of course, only one crank, and if that breaks, the engine is totally disabled; the same happens in the double-crank engine if the after-crank breaks. There is only one connecting-rod, but it is an easy matter to place that beyond suspicion, by making it of ample size and good material; again, the connecting-rod bolts may break, but that argument can be met in the same way, and also by the fact that spare bolts are invariably carried with both forms of engine. If one piston breaks, the other is still capable of working, and the single-crank engine having a flywheel, is better able to propel the ship under such circumstances than the double-crank engine, and is as handy as before for starting and reversing. The link-motion and valve-rods may give way, and so render both cylinders useless, but the chances of the former doing so are a half those of the double-crank engine, while spare parts are not very expensive, and, moreover, are very often carried with both sorts of engines, and so on with all the remaining parts. The working-parts of the single engine, owing to their smaller number, receive more attention, both at sea and in port; and from their unusual size (in order to be symmetrical), some parts are abnormally strong, and so less liable to breakage. The consumption of oil and stores is smaller from the same causes, and their efficiency *at sea* higher, owing to the regulating action of the flywheel. Although the latter did not entirely do away with "racing," it very much reduced it, and enabled a ship to work at *very nearly full speed* in a bad sea with very little waste of power, because the power expended by the engine when the screw is relieved of water is nearly wholly stored in the flywheel, which gives it up again to great advantage when the stern descends, and the screw is deeply buried. The double-engine, when controlled by even the best of governors, *will* "race" somewhat, and power given out then is in great measure lost, owing to the want of *mass* in the working parts; and even if there be no "racing" at all, this is only effected by a reduction of the average power, because the engine is developing its propelling power during the time the screw is immersed, and between those times only sufficient power to overcome internal resistance—in other words, during half the time steam is practically shut off.

It was often urged as a necessary condition that these engines must be "unhandy," but the reverse was the case in actual practice. When properly designed, these engines very seldom stop on the "dead centre," and with a proper starting gear an experienced engineer will never allow it to do so. As the effort of *both* pistons was applied to the crank, if it were ever so little over the centres, the engine would start.

Two-Crank, Two-Cylinder, or Double Engine.—This was the most general form of commercial marine engine, and is the one that with fewest cylinders may have no "dead points," or positions of crank from which it cannot start without special assistance. The cylinders may be of the independent type, in which the steam works expansively, or they may be the high- and low-pressure cylinders of a compound engine.

Two-Crank, Three-Cylinder, Engine is now only of interest from the fact that the earliest triple-expansion engines were designed with a high-pressure cylinder above what was originally the high-pressure cylinder of a compound engine. When of moderate size such engines worked well and took no more room.

Two-Crank, Four-Cylinder, Engines may have one high-pressure and one medium-pressure cylinders mounted on two low-pressure cylinders on the triple-expansion system. The high-pressure and medium-pressure cylinders may be in rear of the low-pressure cylinders, and in line, or nearly so, with them; their piston-rods are connected to the cranks in this case by means of levers, &c. Mr. MacAlpine has adopted this latter design of engine because the momentum of the moving parts of one engine is balanced by that of its neighbour.

Fig. 20 shows a four-cylinder two-crank arrangement, and where fore and aft space is limited such an engine may be used with advantage.

The four cylinders may, however, be on the quadruple principle with one low-pressure only. Most of the early quadruple engines were of this type, the cylinders being arranged as in fig. 20.

Three-Crank Engines.—To obtain a more steady motion at low speeds, and to reduce the *maximum* stress on the crank-shaft, it is necessary to introduce a third cylinder to the diagonal engine, and a third cylinder and third crank to the vertical and horizontal engine. Mr. Scott Russell introduced the three-cylinder oscillating engine for fast paddle steamers, and obtained very good results, although the arrangement was somewhat complicated. Messrs. Maudslay and Field used to make, for the British and foreign navies, many horizontal expansive engines, with three cranks set at angles of 120° with one another, and Messrs. Rennie made some compound horizontal engines, having the two low-pressure cranks opposite one another, and the high-pressure crank midway between them. Several of the Scotch engineers followed the lead of Messrs. John Elder & Co., who, in 1863, fitted a three-crank engine into H.M.S. "Constance," and later on Dr. Kirk, when with that firm, adopted the three-crank vertical engine for ships where the power was such as to require a somewhat unwieldy low-pressure cylinder. The "Constance" was the first ship in the Navy to have compound engines, for on each side of the shaft was a set of cylinders consisting of one high-pressure and two low-pressures. This system not only permits in the compound engine the use of two moderate-sized low-pressure cylinders, but of such an earlier cut-off in them that the "drop" is considerably reduced; the variation in twisting moment is much less than that

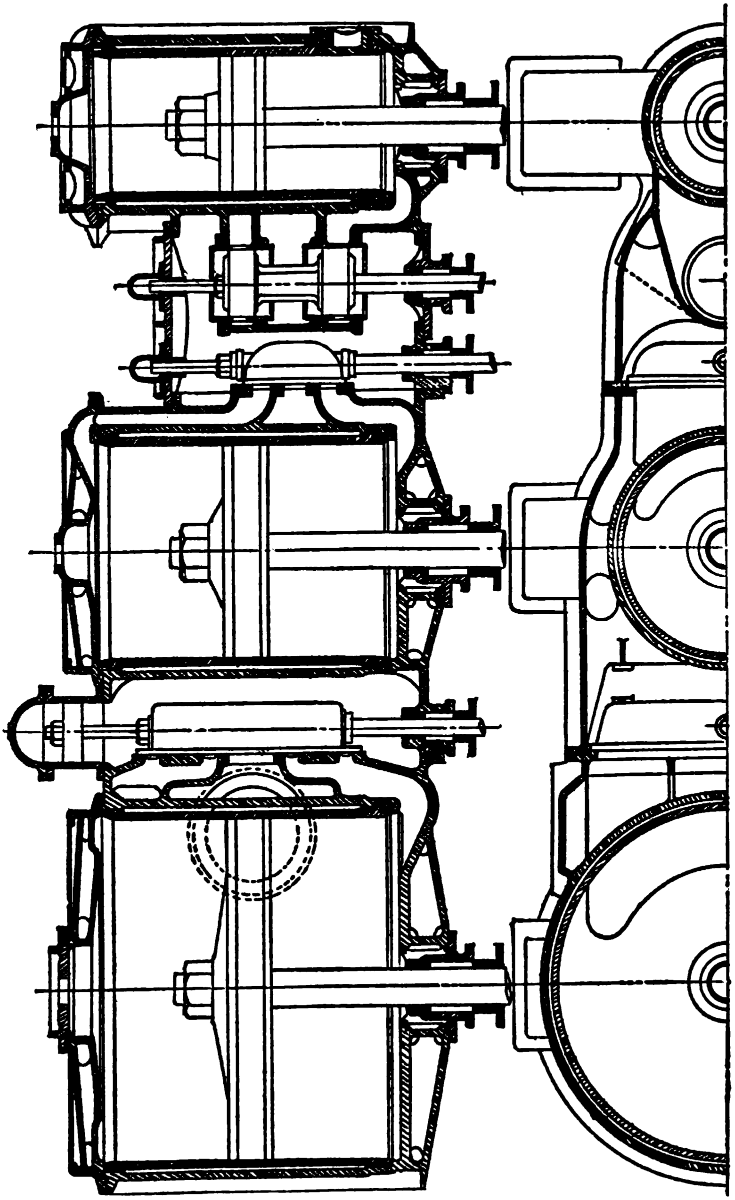

Fig. 19.—Cylinders of a Triple-Expansion Engine.

in the two-crank engine, and the necessity for a large receiver obviated. The modern triple-compound engine is nearly always on the three-crank principle, and consequently enjoys these advantages to enhance its performance. Independently of the advantages of getting steadier motion and reduced stresses on the crank-shaft, there are other reasons which operate in making the three-cylinder arrangement a convenient, if not a necessary one. As a rule, the power of an engine is obtained more efficiently by one cylinder than by two, and by two than by three; but there is a limit of size and weight, beyond which both manufacture and examination when in the ship become extremely expensive and difficult, besides the attendant risk, and it is for this reason that manufacturers are compelled to go beyond the number of two cylinders. Power for power, the three-crank engine took up more room than the two-crank one, and, except in very large sizes, was more expensive to make; but the working parts, piece for piece, are lighter, and more easily made and handled in the ship; and it is of great importance to reduce the *maximum* stresses on the shafts of very large engines, to keep them within the size which may be made with existing appliances, and to give them greater chances of endurance. The cranks are at angles of 120°

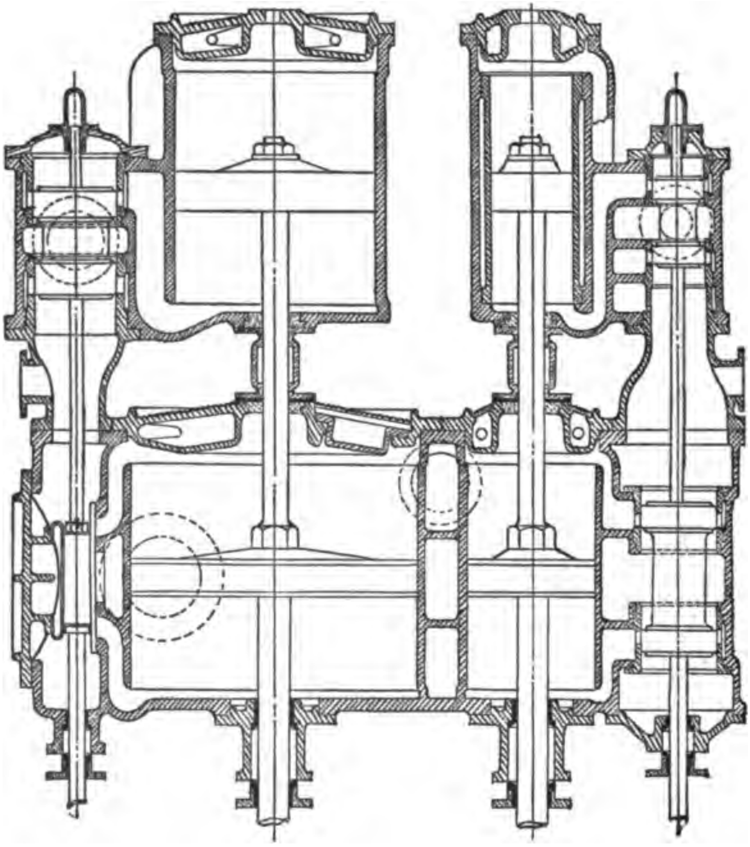

Fig. 20.—Cylinders of a Two-Crank Quadruple Expansion Engine.

with one another in the vertical engine, because the cylinders can be made of such proportions as to admit of this; but in the case of the engines made by Messrs. Rennie, owing to the two piston-rods for the return connection-rod, there was not that choice of ratios of cylinders, and hence the peculiar positions of the cranks, which were so set as to produce the least rational maximum to mean effort. It was usual to place the high-pressure between the two low-pressure cylinders of compound engines, and it was found that this suits the design better than any other.

Three-Crank, Four- and Five-Cylinder, Engines.—By placing a small or high-pressure cylinder over the original high-pressure cylinder of a three-crank compound engine a convenient arrangement for large triple engines was found; but this did not permit of a good distribution of power between the three cranks, and the engine was not so handy as desirable. By fitting two high-pressure cylinders over the two low-pressure cylinders both these difficulties were overcome, and many engines of this kind are doing good work in ocean passenger steamers to-day. By placing a high-pressure over one low-pressure cylinder and a first medium-pressure over the other, and making the middle cylinder the second medium-pressure, a good quadruple-expansion engine is formed, when the power required is so great as to require two low-pressure cylinders.

Three-Crank, Six-Cylinder, Engines may have three low-pressure cylinders on the columns, the high-pressure cylinder over the middle, and a medium-pressure over each of the others, and so form a triple-expansion engine; or the three upper cylinders may be part of a quadruple system.

Four-Crank Engines were originally made by Mr. John Tweedy (Wigham, Richardson & Co.) on the quadruple principle, and are now not uncommon, and made by several firms; but they are, however, better known as triple-expansion engines, having two low-pressure cylinders, one high-pressure and medium-pressure. In this form they were used in the early days of triple-expansion engines by Messrs. Humphrys and Tennant for warship purposes when very large power was required from an engine of light weight, which meant, of course, an engine running at a high number of revolutions, with less risk than is the case with a very large, light low-pressure piston. These engines are now great favourites when high speed is required in both ship and engines, even when the power is not so very great; for by arranging the cranks in a certain and particular way, as invented by Mr. Tweedy and Mr. Otto Schlick, the vibration due to the momentum of the moving parts may be reduced to a minimum without resorting to balance weights or other extraneous means. The four cranks also permit of the engine having the minimum pressure on the crank-journals and bearings.

It is usual and advisable to arrange for the first and second cranks to be opposite one another, and the third and fourth opposite one another, and at right angles to the first and second; there is then no pressure on the bearings between these pairs of cranks, and the pistons moving in opposite direction tend to balance their momentum. The two end cranks being at right angles the couple caused by the momentum of the moving parts, and tending to rock the engine fore and aft, is a minimum, or rather, is less than if opposite one another. By placing the low-pressure cylinders so that their cranks are at right angles with one another the power in them can be raised so that each develops nearly a quarter of the total I.H.P. without having high initial pressures and the consequent high stresses on rods, columns, cranks, &c. If more than four cylinders are necessary for any reason the fifth or sixth may be placed above one or other of the four as in the three-crank engine.

Five-Crank Engines have, so far, been adopted by one firm only—The Central Marine Engine Works—and are the invention of the late Mr. Mudd. Great success is claimed for them, and their owners have been well satisfied with them. The uniformity of turning moment and the balancing of the moving parts they possess, but the fore and aft space taken by them is greater than with the four- or three-crank engine; their cost also is greater. They have a high-pressure, first medium-pressure, second medium-pressure, and two low-pressure cylinders.

Six-Crank Engines are claimed to possess the advantages of the five-crank in a higher degree, and have been fitted in some of the Scout's Class.

Tandem Engines.—The upper cylinders are sometimes supported on a dome-shaped stool, formed with the lower cylinder cover; and sometimes carried on columns which are clear of the lower cover, and independent of it. The latter plan, although not so neat in appearance, admits of the cover of the low-pressure cylinder being raised for the examination of the piston; and by making the cover in halves, bolted together, it may be removed without disturbing the upper cylinder or rod. When the upper cylinder is supported on a stool it is a very common practice to cast this with only one-half of the low-pressure cylinder cover, leaving the other half free for removal to examine the low-pressure piston. In small engines this latter plan does very well, but for large engines the cylinders are apt to rock unless supported by another column; this auxiliary column may, however, be made portable, and the upper cylinders rendered still more stable, by bracing them to one another with tie-rods.

At one time it was a very common practice to compound old expansive engines by placing two small cylinders above the original ones; but although this is a cheaper plan than to provide two new compound cylinders, it was not found to be so economical in working with small engines, owing to the friction of so many glands and pistons. In the plan introduced by Mr. Allibone the high-pressure cylinders were bolted direct to the low-pressure covers, and the high-pressure piston-rod passed through a stuffing-box in the covers on top of the cylinder, and secured in a cross-head, at the outer ends of which were secured the top ends of the two rods belonging to each low-pressure piston. By this plan, however, the number of stuffing-boxes is increased from three for each engine to five, and a greater total height for the engine is required, besides which it is almost impossible to examine the low-pressure cylinder without removing the high-pressure cylinder.

There are several other methods which have been carried out from time to time, in which the tandem cylinders are differently arranged from the foregoing, but none of them have met with sufficient success to induce a repetition of the attempt.

The four-cylinder compound engine was one of the first compound arrangements introduced in the Navy; and, although for some years previous to this the paddle-wheel engines of this type made by the late Mr. John Elder had given very great satisfaction, when this system was applied by Messrs. Humphrys and Tennant to two troop-ships and an ironclad, the results were not so good, and no doubt prevented the general adoption of the compound system for naval purposes for some years.

Variations of Design.—Each manufacturer has a particular method as to details, but with one or two exceptions there is no radical difference in the general design. The principal exceptions to a fixed rule in general arrange-

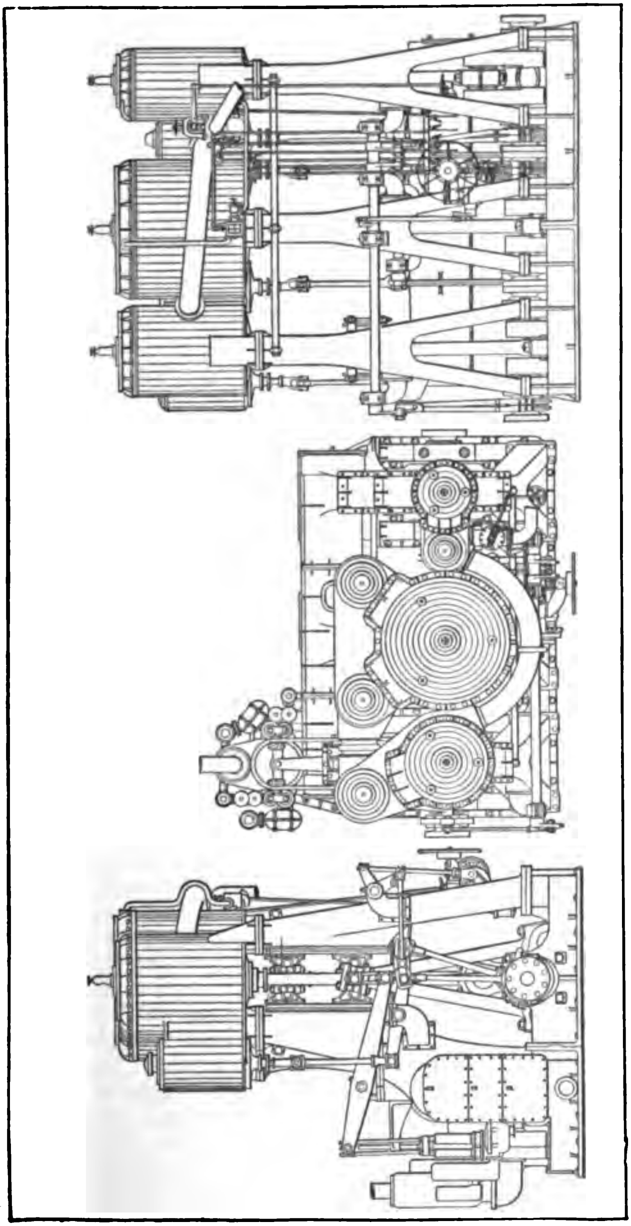

Fig. 21.—Three-Crank Triple-Expansion Engine.

ment are, the position of the condenser and its form, the position of the pumps, and the method of working them, the position of the cylinder valve-boxes, and the position and form of the supporting columns, &c.

The condenser is usually on one side of the engine, and so arranged as to form a base for the columns on that side, and a part of the engine base to which the foundation containing the main bearing is secured; sometimes even the condenser foundation and columns are made in one casting for engines of considerable power. A slight variation from this is effected by some manufacturers, who prefer having a continuous bed-plate without the complicated casting above described, by making the condenser so as to stand on top of the engine foundation-plate. In both these designs the condenser tubes are horizontal, and in a fore and aft direction, and a space is required at one end to draw them for examination, &c.; it is often difficult to provide such a space, and the convenience can only be obtained, in many cases, by taking off a plate from the engine-room bulkhead. To avoid this objection, the condenser is sometimes placed so that the tubes lie horizontally athwartship, and can be drawn between two of the engines, on removal of only a small amount of easily portable gear. When this plan is followed, it is usual for the condenser to be quite independent of the columns, which stand on either side of it with their feet secured direct to the engine foundation-plate. In the case of naval engines the condenser is quite detached from the engines and placed in any convenient position in the engine-room. On the whole, this is a very convenient arrangement, and overcomes other difficulties, besides that of drawing the tubes; for it admits of the engine being erected without the condenser being in place; the condenser may also be made much lighter, and of a simpler and less expensive construction and shape; and, as it forms no integral part of the structure resisting strain, deterioration in the metal from age, galvanic action, &c., does not endanger the engine. Most engineers now prefer the independent condenser for mercantile engines in all but the smallest engines, making it of cylindrical form, with cast-iron water ends and doors; the body is made of wrought iron or steel when weight is a consideration, and of cast iron when it is not. Setting aside the cost of the pattern, the latter is much the cheaper arrangement, and even with the pattern generally it may be made for less money as a casting. The condensers for small craft, as torpedo boats and for "destroyers," the body and even the ends are of sheet copper or sheet brass.

If the condenser is placed on one side of the engine, the centre of gravity of the machine will be out of the centre line of the ship, unless it can be balanced by some other part. As this balancing seldom happens in ships which would be seriously affected by such a condition, some engineers have rectified the defect by moving the condenser as above described, until it is under the cylinders and central with them, the tubes being athwartship. This, again, necessitates the removal of all gearing from the middle of the engines, and as the cylinder valve-boxes are "outside," the working parts of one engine are hid from the view of the engineer when he is attending to the other. This has been obviated by making another removal of the condenser to the aft end of the engine, where it forms a support for the cylinder, and does not interfere with the position or view of the working parts. But whether the condenser be in the middle of the engine, or at the aft end, so long as it is central, and above the crank-shaft, it forms a very serious obstacle when this very important working part requires removal for examination or through defects, and at all times it is in the way of the main bearings which require attention. To have a condenser which shall be central, and yet form no obstruction, Sir William Allan patented an arrangement, in which it is *below* the shaft with the tubes athwartships, and forming a part of the engine foundation. The objection to the plan is the great height of shaft-centre necessitated by it; in addition to which, the change of form caused by the different temperatures of the condenser distorts the foundation, and puts the shaft-bearings *out of truth*, and that the bilge-water on the hot-condenser sides and bottom seriously corrodes it, besides causing foul smells in the engine-room; the exhaust-pipe, too, is of necessity longer than usual, and the discharge-pipes from all the pumps will have the same objectionable feature.

The condenser is sometimes arranged with the tubes vertical, and is then generally cylindrical, and has a column cast with it, so as to form a part of the engine framework; by this means, a cheap and compact condenser is formed, and one that admits of the tubes being drawn, when no other kind of design could be so arranged. This plan is especially favourable for yachts having good power, without a large amount of breadth of beam.

In large steamers, it is of very little consequence if the centre of gravity of the machinery is a little out of the centre line; and in those of moderate size, the condenser can always be balanced by some other fixed weight, such as the donkey-boiler; while, in small steamers, this can be effected by placing the boiler a very little way out of the centre line, on the side opposite to that of the condenser. Under these circumstances, it is not surprising that most engineers still design their engines with the condenser on one side.

The position of the pumps naturally depends very much on that of the condenser, and the method of working the pumps is not, as sometimes supposed, optional, but consequent on the position of these relative to the cylinders.

When the condenser is on one side of the engine, it is almost a necessity to have the pumps behind it; and the only convenient way of working them is then by means of levers, which obtain their motion from the piston-rod crosshead. As the condenser so placed is the most general arrangement, so the generality of manufacturers work the pumps by means of levers, the pumps being behind the condenser.

This is by far the most convenient method, as by it the pumps (which really require very slight attention compared with that bestowed on the shaft-bearings, rods, link-motion, &c.), are out of the way of all the working parts, and in that place on the engine where they are most easily examined, and also nearer to the inlet and discharge-valves, than they would be in any other position. The weight of the moving parts of the pumps helps to balance the weight of the pistons and rods, and although this is not of much consequence when the engine is running fast, it is very important when starting or moving slowly, and must at all times tend to make the engine run steadily. In the case of high-speed engines, which must be balanced to avoid excessive vibration, the pumps assist materially when moving in the opposite direction to that of the piston, from which they receive their motion.

When the condenser is placed in the middle of the engine, although not an absolute necessity, it is certainly by far the most convenient arrangement to have the pumps central, or nearly so, and worked *direct* from the piston crosshead; hence, it is usual to find in all *designs* of the kind, the pumps worked direct, and situated about the centre line of the engine. The objections to this plan are always of moment; but with long-stroke engines they become almost insurmountable. The pumps are always in the way of the attendants, and as the arms of the piston-rod crosshead are usually long, so as to keep the pumps as far away from the crank-shaft as possible, it is dangerous to approach the centre of the engine for fear of being guillotined. When the stroke of the engine is long compared with the diameter of the cylinders, it is impossible to get sufficient area through the buckets of the air and circulating pumps when single-acting, and at high speeds the concussion from the water is very great. The weight of the pumps and the crosshead is added to that of the piston, so that the downward velocity of the latter is naturally greater than that upward; and although this can be remedied to some extent by adjusting the steam supply, so that the engine runs steadily at full speed, there must always remain some slight difficulty at starting. As a set-off to this, the cost and friction of the levers and links are saved, but this is scarcely sufficient to counterbalance objections so serious as the above.

The pumps are sometimes worked by an independent engine, and may then be placed in any convenient position; but, in addition to the cost of such an engine, there is the objection that much room is occupied by it, and it is doubtful whether more power is not expended in doing the work, than would be the case if the pumps were worked in the usual way; and in any case the power taken from the main engine is developed more economically than that from a small auxiliary engine, even if compound. They can, however, be worked at that speed suited to the exact requirements.

In the case of expansive engines, it was usual to arrange the cylinders so that both valve faces came between them in such a way as to form a valve-box common to the two; this was a very convenient plan, and economical both in cost and space. It involved, however, the necessity of four bearings to the crank-shaft, and where the engines were small, it was no easy matter to do any work to the valve-faces when required. For these reasons manufacturers made them sometimes with the valve-faces outwards, as is usual with horizontal engines. When compound engines were first introduced, each maker followed the arrangement to which he was accustomed in expansive engines; but it was found that neither of the above plans permitted of a sufficient space for a receiver when the cranks were placed at right angles, and both had serious defects. When the faces were "outside," the exhaust-steam from the high-pressure had to traverse a comparatively long pipe to reach the low-pressure, and for the small volume of this pipe a very large surface was exposed to loss from radiation. When the faces were inside, and "flat"-valves used, there could not be a valve-box common to both, but each box was shut off from the other, and the temperature of one very much higher than that of the other; the consequence was, that the engine centres were very much spread out, and the castings about the valve-boxes were often found to be cracked. This latter defect has, by care in design, been remedied, but there still remain the wide centres and the liability to leakage of high-pressure steam to the low-pressure valve-box without detection, except by the indicator, or the abnormal pressure in the receiver when the engines are "standing." By using a piston-valve, however, for the high-pressure cylinder, taking the steam on the inside (*vide* fig. 19), and a flat-valve, or even a piston-valve, taking steam on the outside for the next cylinder, one valve-chest common to the two is possible and convenient.

To avoid the evils above mentioned, and to obtain a good receiver, most manufacturers place the valve-box of the high-pressure cylinder "outwards," and that of the medium and low-pressure between the cylinders, and provide a belt around the high-pressure one for the steam to exhaust from it to the medium-pressure, and, similarly, from the medium to the low-pressure cylinder. When the space between the cylinders is not sufficient for a receiver, this belt is made the whole length of the cylinder, and in this way space is provided. The medium-pressure and low-pressure valve-box covers are on top (in vertical engines), while that of the high is on front, and sometimes is provided with one on top as well when a flat-valve is used, and on top only when a piston-valve is used. Peep-holes with suitable covers are formed in the sides of valve-boxes of large engines, which permit of the valves being sighted without the labour and time required to remove the box covers.

In order to reduce the space occupied by the cylinders, and likewise to suit the requirements of certain special valve motions, the valve-boxes of both cylinders are placed on the front side of the cylinders, so that the faces are in a plane parallel to the keel. When this is so, the cylinders may be quite close together, and no space is required at either end of the engine-room casing for valve-boxes or to remove valve-doors; the cranks are also arranged to come closer together, to suit the reduced centres of cylinders, by doing away with one of the middle bearings and the room for eccentrics. The only difficulty with this arrangement is to get space for a receiver in the case of compound engines, and for the passage of the exhaust steam from the low-pressure cylinder to the condenser. The adoption of the three-crank triple-compound engine has developed a very large amount of skill and ingenuity in the arranging of both cylinders and valve-boxes, so as to minimise the length of engine.

The supporting columns differ in form and position as well as in material. The back columns, or those carrying the guides, are, as a rule, of cast iron, and of rectangular section, to suit the requirements. In naval engines they are often of cast steel. The front columns, or those on the "starting" side, are seldom exactly the same with any two makers, and sometimes widely different. They may be cast iron and of hollow rectangular section (fig. 21). H section (fig. 12), or hollow cylindrical, or of wrought iron or steel, of circular section. When the latter, there are generally for large engines, one in wake of each main bearing; and two for small compound engines, one at each corner, and for three-crank engines of small size three. The same rule applies to circular section cast-iron columns with some manufacturers. Rectangular section cast-iron columns are sometimes substituted for circular ones, when arranged in the same way as the circular section columns, to be in keeping with those at the back. Rectangular section columns for large engines, and H section for small engines, are generally placed opposite the back columns, and have guides on them in the same way as these latter. The guides on the back columns are usually the "ahead-going" ones; but as they are generally placed on the condenser, and, therefore, apt to become warm, some engineers make those on the front columns do this duty by fitting a screw propeller of reverse thread. Columns, both back and front, are generally made to approach straight lines in outline, although some still have what are supposed to be graceful curves.

CHAPTER V.

ENGINES, SIMPLE AND COMPOUND.

Elementary Steam-Engine.—The steam-engine, in its most elementary form, has only one cylinder, into which the steam is admitted at each end alternately, so as to move the piston backwards and forwards, and having performed its work is then allowed to escape into the atmosphere. Although from certain causes this was not the form of steam-engine as first invented, it is nevertheless the most simple one, and by taking it as the origin, the genesis of the steam-engine can be better explained. As the successful engines for marine purposes are largely those having cylinders and pistons, it will be unnecessary in this chapter to deal with any other forms.

Genesis of the Compound Engine.—The exhaust steam issuing from an engine having a late cut-off and an initial pressure of, say, two atmospheres (or 15 lbs. above atmospheric pressure), would attract the attention of an observant engineer from the force with which it emerges from the exhaust-pipe, and would naturally lead him to inquire how so great a waste of energy *might* be avoided. It would be clear to him that there was sufficient remaining in it to do useful duty after it had accomplished its work in the cylinder. If he were acquainted with the steam-engine of Watt, he would probably suggest that, instead of allowing it to escape into the atmosphere, it might be conducted to the cylinder of a condensing engine, which could work with a steam pressure of one atmosphere, and while propelling the piston of this second engine, would not cause any more back pressure in the first cylinder than before. Such an arrangement would be a combination of a high-pressure and a condensing engine, and hence we find it called a *compound engine.*

This combination has been carried out in some instances in modern times; and no doubt it was from his observation of Trevithick's high-pressure non-condensing engine, together with his knowledge of Watt's condensing engine, that Woolf was induced to introduce the "compound" engine.

The idea of a compound engine, however, was due to the genius of Hornblower, a Cornish engineer, who in 1781 took out a patent, in which he claimed:—"I use two steam vessels, in which the steam is to act, and which in other steam-engines are called cylinders. I employ the steam, after it has acted in the first vessel, to operate a second time in the other, by permiting it to expand itself, which I do by connecting the vessels together, and forming proper channels and apertures whereby the steam shall, occasionally, go in and out of the said vessels, &c."

Arthur Woolf, in his patent, taken out in 1804, states that "if the engine be constructed originally with the intention of adopting my said improvement, it ought to have two steam vessels of different dimensions, according to the temperature or the expansive force determined to be communicated to the steam made use of in working the engine; for the smaller steam vessel or cylinder must be the measure of the larger. . . . The small cylinder should have a communication, both at its top and bottom, with the boiler which supplies the steam. . . . The top of the small cylinder should have a communication with the bottom of the larger cylinder, and the bottom of the smaller one with the top of the larger, with proper means to open and shut those alternately by cocks or valves, &c., . . . and both top and bottom of the large cylinder should, while the engine is at work, communicate alternately with the condensing vessel." He proposed to use steam at a pressure of 40 lbs.

Expansive Engine.—If the observer, however, happened to be better acquainted with the expansive force of steam than with the use of a condenser, he would suggest that the steam should be cut off at such an earlier part of the stroke as would ensure its pressure, at emission, being only slightly above that of the atmosphere. Such an engine would naturally be called *expansive*, in contradistinction to the original engine, working without expansion. Any further attempt at increased expansion would prove fruitless, as the steam, expanded below the pressure of the atmosphere, will fail to escape into it when opened to exhaust; besides which, the "back" pressure on the other side of the piston would, during the latter part of the stroke, be greater than the forward. Only by connecting the exhaust pipe to the condenser, in which the pressure would be 10 or 12 lbs. below that of the atmosphere, could a higher rate of expansion be obtained.

All the very early marine engines were *expansive*, and, according to circumstances, either condensing or non-condensing, but generally the former. This condition was to a very large extent imposed on makers of engines, because in those days the boilers could not be economically made for pressures much above that of the atmosphere, even for small steamers, and for large ones it was an impossibility. With a better knowledge of boiler-making, and a supply of larger and thicker plates, engineers began to attempt to work their engines with steam of a higher pressure, and obtained thereby satisfactory results. A few accidents, however—generally caused by the ignorance and carelessness of the attendants—tended to check any very great advance in this direction, and it was not until public confidence was regained that engineers could continue the movement to any great extent.

Effects of Increase of Pressure.—If an engine is to work economically, so far as steam is concerned, it has been stated that the terminal pressure, before admission to the condenser, should be as low as possible consistent with good working.

It is generally easy to maintain a vacuum of 26 inches in a modern condenser, and 24 inches was usual with even a poor form of jet condenser. But as some engineers prefer to work their engines with only 24 to 25 inches of vacuum (for the sake of obtaining warm feed-water), let it be assumed that 24 inches is the vacuum in the condenser. In a well-designed engine, the vacuum in the cylinder, after exhaust, should be at least within 2 inches of this, which means that the back pressure is 4 lbs., corresponding to 22 inches of vacuum. When the full benefit of expansion is required, the terminal pressure should not exceed 7 lbs., which will be 3 lbs. above the back pressure.

In order to appreciate fully what is encountered in making advances in boiler pressures, it will be well to compare two engines working under the conditions set out above.

Suppose these two engines to have each one cylinder of the same diameter and stroke, the boiler pressure of the first to be 2 atmospheres, or 30 lbs. absolute, that of the second 3 atmospheres, or 45 lbs. absolute, the terminal pressure in both cases to be 7 lbs., and the back pressure 4 lbs.—The cut-off in the first will be $\frac{7}{30}$ of the stroke, or a rate of expansion of 4·285, and the mean pressure with an initial of 30 lbs. is 17·4 lbs.; deducting 4 lbs. of back pressure, the *effective* mean pressure will be 13·4 lbs. In the second engine, the cut-off will be $\frac{7}{45}$ of the stroke, and the rate of expansion 6·43, and the mean pressure with an initial of 45 lbs. is 20 lbs.; deducting, as before, 4 lbs. for back pressure, the *effective* mean pressure is 16 lbs. The *effective initial* pressures will be 30 − 4, or 26 lbs., and 45 − 4, or 41 lbs., respectively.

Since the cylinders are of the same capacity, and the terminal pressures are the same, each engine consumes the same *weight* of steam; but since the total heat of evaporation of steam from the temperature corresponding to 4 lbs., and at that corresponding to 45 lbs., is 1130° Fah., while at that corresponding to 30 lbs. it is 1121° Fah., there will be an expenditure of fuel to obtain the steam at 45 lbs. slightly in excess of that at 30 lbs. As this, however, amounts to less than 1 per cent., it may be neglected, and the cost of the steam assumed to be the same. It will be seen then that, with the advance of boiler pressure, there is an advance in mean pressure, and the gain in power amounts to nearly 20 per cent.; but the initial load on the piston of this more economic engine is 57 per cent. higher than that on the more wasteful one, and consequently the rods, framing, &c., must be 57 per cent. stronger; the shaft will be increased in size, and the cylinder and passages must be stronger. Altogether, then, the engine will become heavier and more costly, as the boiler pressure is increased.

To render the comparison strictly fair, it would be necessary to take two engines of *equal power*, so that if the stroke of the pistons is the same, their areas will be inversely proportioned to the mean pressures, and, consequently, the initial strains will now be 26 lbs. and $\frac{13 \cdot 4}{16}$ of 41, or 34·5 lbs., which gives an excess of 31·3 per cent.

Progress Made by Early Marine Engineers.—The early marine engineers, however, did not advance on these lines as a rule, for very nearly the same rate of expansion was observed with steam of three atmospheres as had obtained with steam of atmospheric pressure, and consequently very little benefit was derived in economy, compared with what might have been the case had they done so. The chief result accruing from the increased boiler pressure was the larger indicated horse-power developed by engines of the same size as formerly, partly due to the augmentation of mean pressures, and partly to the increased piston speed resulting from these.

Engines of as much as 200 N.H.P., working with steam of 60 lbs. pressure, were supplied to the Navy by Messrs. Penn and Messrs. Maudslay as early as 1853; but a period of more than fifteen years elapsed before the Admiralty again employed such pressure in any larger ship than a gunboat.

Just previous to the re-introduction of this high boiler pressure into the Navy, warships were being supplied with boilers loaded to 30 lbs. per square inch; the rate of expansion was about 1·5 when at full speed, and about 3 when cruising. Some merchant ships were still working under similar conditions; but the majority of new vessels, unless fitted with compound engines, were supplied with boilers loaded to 40 or 50 lbs. per square inch, and the rate of expansion was about 3 to 4 when on regular service. The advocates of the expansive system (in order to rival the performance of the compound engine) used a boiler pressure of 60 lbs., and a rate of expansion of 5. Some small steamers, and some larger ones in which economy was sacrificed to speed, had engines supplied with steam of 5 atmospheres and a rate of expansion of only 1·5.

A comparison of four such engines, having the same stroke and developing the same power, can be made from the following results:—

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TABLE VII.

Load on safety valves,	lbs.	30	45	60	60
Diameter of cylinder,	ins.	50	50·8	51·1	38
Initial absolute pressure,	lbs.	45	60	75	75
Cut-off,	stroke	$\frac{6}{10}$	$\frac{3}{10}$	$\frac{2}{10}$	$\frac{6}{10}$
Back pressure,	lbs.	4	4	4	4
Mean effective pressure,	”	36·77	35·66	35·15	63·95
Terminal ”	”	27·00	18·00	15·00	45·00
Maximum load on piston,	”	80,483	113,456	145,550	80,514
Mean ” ”	”	72,180	72,180	72,180	72,180
Ratio of max. to mean,	”	1·115	1·572	2·016	1·115
Weight of steam used,	”	893	609	515	868

Examples (1), (2), and (3) show conclusively that by increasing the boiler pressure and rate of expansion, so as to obtain nearly the same mean pressure, the weight of steam used is considerably diminished and the terminal pressure considerably reduced, but that there is an increase in the maximum load on the piston proportionate to the *absolute* pressure in the boiler, so that the ratio of maximum to mean pressure of example (3) exceeds that of example (1) by more than 80 per cent.

Although such an increase in weight of machinery as would be necessitated by so great an increase in load, may not be of much importance in some ships, in others it would be prohibitive; for when the power required for certain speeds of ship becomes large compared with the displacement, it requires the utmost care in design to keep down the weight, so as to admit of the engines being carried by the ship on the required draught of water. For this reason, in actual practice, it was found impossible to use steam of over 50 lbs. pressure (above the atmosphere) in very fast river steamers, or even in high-powered steamers for Channel service of moderate size, on account of the limited speed of piston obtainable with the paddle-wheel. By means of forced draught, constructing the ship and engines as much as possible of steel, and a special design of light compound engines, pressures of 100 to 125 lbs. are now employed in such ships, and in some few cases triple-expansion engines using steam of 175 lbs. pressure have been fitted in paddle steamers of high speed. With the screw engine, where the number of revolutions per minute may be largely varied without any very widely differing results, and which may have a long stroke of piston without encroaching too much on passenger or cargo space, the evils arising from high initial pressures in non-compound engines may be so mitigated as to permit of their being used for the fast navigation of shallow waters by comparatively small steamers; but it seldom happens that such ships have sufficient draught of water to admit of sufficiently large screws for efficient working.

Example (4) is given that the effect of two widely different boiler pressures may be compared, when the rate of expansion is the same. The mean pressure is, of course, much higher, and, but for the back pressure being the same, would bear the same proportion to that at the boiler pressure of 30 lbs. as the initial absolute pressures, viz., 5 to 3. The weight of steam used is very little less than that of 30 lbs. pressure, and, owing to the reduction in the size of the piston, the maximum load in this case is practically the same as in example (1). The pressure at exhaust is exceedingly high, being 45 lbs. absolute, or 30 lbs. above that of the atmosphere, so that it is capable of doing considerably more work, if admitted into another cylinder of larger size, than that of the first; and even if admitted into one of the same size (provided it finally exhausts into a condenser), more work will be obtained from the steam than if it is allowed simply to escape into the atmosphere. No advantage, however, will be gained by using a second cylinder of the same size over that of exhausting the steam direct into a condenser; in fact, in practice, from the resistance by friction, &c., a second cylinder would give a positive loss.

Receiver Compound Engine.—Now, suppose that an engine working under the conditions set out in example (4) (so far as pressure and cut-off are concerned) exhausts into a steam-tight space, so that there is back pressure in front of the piston equal to the pressure in this receiver of the exhaust steam; and suppose, further, that the steam is taken away from the receiver at the same rate as it is supplied by the cylinder, there will then be a constant mean pressure maintained there. For the sake of fixing the application to example (4), suppose, again, the pressure in its receiver to be 30 lbs. *absolute*, then the mean pressure in the cylinder will be $67 \cdot 95 - 30 = 37 \cdot 95$ lbs. only. Now, suppose a second and larger cylinder to be supplied with steam from the receiver at such a rate that there is no change of pressure in it (this being accomplished by so arranging the cut-off in the second cylinder, that the weight of steam taken by it equals the weight of steam exhausted from the first one); the cut-off may be determined from the formula $p\,v =$ const.; so that if V be the volume of the second cylinder and v that of the first, 45 lbs. the terminal pressure in the small and 30 lbs. the initial in the large cylinder. Then, cut-off in the second cylinder $= \frac{45\,v}{30\,V} = \frac{3}{2} \times \frac{v}{V}$, and if the ratio of V to v is 3, the cut-off in the second cylinder is ½ stroke, and the rate of expansion in it 2.

The mean pressure, with an initial pressure of 30 lbs. and a rate of expansion 2, is 25·38 lbs.; and allowing for a back pressure of 4 lbs., the mean effective pressure in the second cylinder is 21·38 lbs.

Since the area of the second piston is three times that of the first, the work done in the second cylinder is equivalent to what might be done by one of the same area as the first, with a mean effective pressure three times as great, or 64·14 lbs. per square inch. It will be seen from this that the total work done by the combined cylinders is the same as would be done by the original cylinder with a pressure of 37·95 + 64·14, or 102·09 lbs. per square inch; hence we find that there is a gain of nearly 60 per cent. from the introduction of the second cylinder. So far, the compound engine would be undoubtedly more economical than the expansive engine, as exemplified in examples (1), (2), and (4), but less in this particular instance than example (3).

Expansive and Compound Engines Compared.—To examine the relative economy of a compound and of a simple expansive engine, it is necessary that they should both work with the same boiler pressure and the same rate of expansion. Now examples (3) and (4) satisfied the first condition, and if the second is also satisfied, then they may be compared. The rate of expansion in example (4) is 1·666, and since the volume of steam in the second cylinder at the end of its stroke will be three times that in the first at the same period, the total expansion effected by both cylinders will be 3 × 1·666, or five times. The cut-off in example (3) was two-tenths the stroke, and therefore its rate of expansion is five; so that these two examples may be compared as to the efficiency of the steam. The effective pressure of the compound system may be referred to the large cylinder, in the same way in which it was referred to the small one, and will be that actually on the large cylinder, together with that on the small one *divided* by the ratio of their capacities; hence, effective mean pressure referred to the large cylinder is $21{\cdot}38 + \frac{37{\cdot}95}{3}$, or 34·03 lbs. per square inch. It will be seen that this is 1·12 lbs. less than that obtained in the simple expansive engine, and therefore a loss has occurred in the compound system.

Suppose, now, that the cut-off in the large cylinder is so altered that the pressure in the receiver is 45 lbs., so that it receives steam at the same pressure as that which exhausts from the small one; in this case there will be no "drop" in the pressure from commencement of exhaust to the end in the small cylinder.

The mean effective pressure in the small cylinder is now 67·95 – 45, or 22·95 lbs. per square inch.

The cut-off in the large cylinder $= \frac{45\,v}{45\,V}$, or $\frac{1}{3}$ the stroke, which gives 3 as the rate of expansion.

With an initial pressure of 45 lbs. and rate of expansion 3, the mean pressure is 31·5 lbs.; allowing 4 lbs. for back pressure, the effective mean pressure is 27·5 lbs.

Referred to the large cylinder the effective mean pressure of the system is now $27{\cdot}5 + \frac{22{\cdot}95}{3}$, or 35·15 lbs. on the square inch, or *exactly the same* as that obtained in the simple expansive engine.

Effect of "Drop" in the Receiver.—It is seen from the above, then, when no "drop" occurs there is no loss of efficiency; but that when the pressure in the receiver is less than the terminal pressure in the small cylinder, there is a loss of effective mean pressure. This arises from the steam being allowed to *expand* from the small cylinder into the receiver *without doing work*. But it is known that, when this takes place, the steam becomes superheated; for, inasmuch as the loss of pressure has occurred without conversion into external work or loss of heat in any way, it must appear in some other form. Although this loss is not wholly recovered, it is considerably reduced by the benefit which the steam derives from the superheating in expanding in the large cylinder, and it is therefore more apparent than real.

Division of the Work.—It will be also seen that, as the ratio of the cylinders' capacity is 3, and the effective mean pressures 22·95 lbs. and 27·5 lbs., the work done in the small cylinder to that in the large is as 22·95 to 27·5 × 3, or nearly 1 to 3·6; while in the former case it was as 37·95 to 21·38 × 3, or nearly 1 to 1·7.

Therefore, with an *earlier* cut-off in the large cylinder, *more work* is developed in it than is the case with a later cut-off; and, moreover, with this ratio of cylinders, in order to get the highest efficiency of the steam, the ratio of the work done is as 1 to 3·6; and the initial pressure on the large piston is 41, and on the small $\frac{75 - 45}{3}$, or 10, as against 71 on the expansion engine of *equal* size; and even if the compound engine were arranged with one cylinder above the other, the combined initial pressure would be 41 + 10, or 51, as against 71 of the simple expansive.

Direct Expansion Compound Engine.—The compound engine may, however, work without an intermediate receiver, if the pistons are arranged to move simultaneously, either in the same or opposite directions. To consider this case—suppose the cylinders to be side by side, and the pistons to move in opposite directions, as proposed by Woolf, so that when the small piston has *receded* one-tenth of the stroke, the large one has advanced by exactly the same amount, and the space between them is $0·9\,v + 0·1\,V$; the volume of steam at commencement of exhaust is v, and the pressure at that period, as before, 45 lbs.; the volume at any point of the stroke $\left(\frac{n}{10}\right)$, or the space

$$\text{between the pistons} = \frac{n}{10}V + \frac{10 - n}{10}v,$$

$$= \frac{n}{10}(V - v) + v,$$

and since $V = 3\,v$, space between the piston at n-tenths of the stroke $= v\left(\frac{n}{5} + 1\right)$.

The pressure at this point $= 45 \div \left(\frac{n}{5} + 1\right)$. The pressures at every tenth of the stroke from 0 to 10 will be 45, 37·5, 32·14, 28·12, 25·0, 22·5, 20·45, 18·75, 17·3, 16·07, 15·0; the mean of which is 24·78 lbs. per square inch. Deduct this from 67·95, and the effective mean pressure in the small cylinder is 43·17; deduct 4 lbs. from 24·78 lbs., and the effective mean pressure in the large cylinder is 20·78 lbs.

The effective mean pressure of this system, referred to the large cylinder, is now $20·78 + \frac{43·17}{3}$, or 35·17 lbs. per square inch, which is the same as that obtained by direct expansion in one cylinder—example (3).

It will be observed, however, that the ratio of the power exerted by the small cylinder to that exerted by the large one, is as 43·17 to 3 × 20·78, or 1 to 1·44, being nearer an equal distribution of the work than in either of the cases of the intermediate receiver engine. This latter result is caused principally by the decreased back pressure in the small cylinder.

In actual practice there are certain causes which materially modify the results shown by both of these forms of compound engine, as illustrated in the foregoing.

In the receiver compound engine, the pressure in the receiver is not constant, because of its limited size, the difference in the periods of exhaust and admission, and the cushioning after cut-off in the large cylinder by the small piston.

The direct expansion engine is only nominally without a receiver, as the space between the small piston and the large one is often considerable (*v.* fig. 20), from the size of the communication pipes and the valve-box of the large cylinder. The valve of the large cylinder cuts off some time before the small one ceases to exhaust, causing cushioning in the latter and in the spaces; the small cylinder also commences to exhaust before the large one can take steam.

Direct expansion compound cylinders have, however, been arranged so that one cylinder communicates directly with the other, without any intervening space, by placing the cylinders side by side, and causing the pistons to operate on cranks set opposite one another (*v.* fig. 126). But such engines have, for other reasons, proved generally unsuitable for propelling.

Requisites in the Marine Engine.—A marine engine must be (1) readily started and reversed; (2) it must have a turning moment or torsion as nearly uniform as possible; (3) it must be able to work continuously for long periods without stoppage from any cause; and (4) and lastly, it must be economical.

The first condition is a *sine quâ non*, and is generally fulfilled by having two or more cylinders, operating on cranks at suitable angles.

The second condition is absolutely necessary when weight is a serious consideration, and is very fairly satisfied by the two or more cranks at proper angles, and the cylinders so designed as to divide the work equally between them. When weight and size of working parts are of small importance, a single-crank engine, or an engine with two cranks opposite one another, each having a flywheel to ensure uniformity of motion, will answer.

The third condition will depend very much on the *variation* in stress, so that the engine with small initial pressure compared with mean pressure, is more likely to fulfil it than one with large initial pressures.

The fourth condition, which is of the utmost importance to the shipowner, is very well complied with by the receiver form of compound engine, and better still by the triple and quadruple compound engines.

It has been shown that a compound engine in which the steam expands direct from the one cylinder to the other, and the receiver engine having a cut-off in the large cylinder the same as the ratio of the small to the large cylinder, are both equal to the direct expansion simple engine in efficiency of the steam. But neither of these two forms answers conditions (1) and (2), and it is doubtful if they do condition (3), while experience has shown that they do not always satisfy condition (4). Although numerous attempts were made by very many able men to design and construct an engine on the compound system which should have the highest efficiency, so far as steam used is concerned, the fact remains, that the form which is theoretically least efficient was the one that survived, and was recognised on all sides as the best compromise of the various conditions governing the choice of a marine engine.

With two cylinders and cranks at right angles, there must inevitably be some amount of "drop," if the work is to be evenly divided when the power is so much as is usually developed by a marine engine at service speed. The crank of the small cylinder should lead—that is, should be in advance of the other crank by 90°, or such other angle as it is deemed best to set the cranks at. When this is the case, the small cylinder begins to exhaust just after the crank of the large cylinder has got well over the centre, and tends to maintain a constant pressure on the large piston through the earlier portion of its stroke, and at cut-off the pressure in the receiver is not much below its average pressure. If, on the other hand, the crank of the large cylinder leads, exhaust takes place only a little before cut-off in the large cylinder, and causes a hump in the indicator-diagram, showing an increase in the amount of "drop," and that with no diminishing in the mean back pressure in the small cylinder. Engines having the low-pressure engine crank as the leading one are also generally unhandy. The first triple-compound engines were, as a rule, designed so that the high-pressure crank "led," or was in advance of the medium-pressure crank by 120°, and the medium-pressure crank 120° in advance of the low-pressure crank; but in modern practice it is usual to fit the low-pressure crank in advance of the medium-pressure crank, &c., as in this way there is less variation in temperature although the load on the pistons is less during the first half of the stroke and greater during the second half than is the case when the high-pressure crank leads, and the engine is not appreciably less "handy."

The first cylinder of a compound engine is called the "high" pressure, and the last the "low," from their association with the condensing and non-condensing engine. For convenience in speaking of them, they are designated by the initials H.P. and L.P. Hitherto, in the chapter, all comparisons of the compound with the simple expansive engine have been made on the supposition that the expansive engine has only one cylinder of the same capacity as the low-pressure one of the compound system. To render the comparison perfectly fair, it will be necessary to take such cases as may be found in actual practice. In triple-expansion engines the middle cylinder is called the "medium-pressure," and sometimes the "intermediate," and designated by the initials M.P.; and in quadruple engines the third cylinder is called the 2nd M.P.

Comparative Efficiency of the Various Kinds of Marine Engine.—(1) *A single-cylinder expansive engine:* rate of expansion, 5; initial pressure, 80 lbs. absolute; area of piston, A.

Mean pressure - - - = 41·76 lbs.
Effective mean pressure - = 41·76 - 4 = 37·76 lbs.
Effective initial load on piston = (80 - 4) A = 76 A lbs.
Efficiency of the system - = 1·00.

(2) *A single-crank compound engine:* rate of expansion, 5; initial pressure, 80 lbs. absolute; area, L.P. piston, A; ratio of cylinders, R, generally in practice, 4.

Effective mean pressure referred to L.P. piston = 41·76 - 4 = 37·76 lbs.

Terminal pressure in H.P., and initial pressure in L.P. $= \frac{R}{5} \times 80$
$= 64$ lbs.

Effective initial load on H.P. piston $= (80 - 64)\frac{A}{4} = 4$ A.

„ „ L.P. „ = (64 - 4) A = 60 A.
Efficiency of the system - - - = 1·00.

Total load on crank is therefore 64 A, against 76 A with the simple expansive engine.

As in actual practice there is invariably a drop, which will amount to as much as 10 lbs. in an engine of this kind, the initial pressure in the low-pressure cylinders being decreased by that amount, and that of the high-pressure increased. So that, actually, the total loads will be as 56·5 to 76, or a saving in the compound engine of over 25 per cent. of the strain put on the rods, framing, &c.; and also enabling a large reduction to be made in the diameter of the shafting. The engine will work much more steadily, owing to the ratio of maximum to mean pressure being so largely reduced; and the handiness very much increased from the cut-off in the high-pressure cylinder being so late as $\frac{8}{10}$ the stroke. The friction of the two cylinders, &c., is considerably greater than that of the one, but this is more than balanced by the reduction in friction on the guides and journals; and the friction on the valve of the single cylinder exposed to high-pressure steam will be more than the combined friction of the two valves, the small one of which only is so exposed, while the expansion valve (which is necessary to the single cylinder for so early a cut-off) will also increase its loss from friction.

The compound engine compares more favourably with the simple expansive when both have two cylinders and two cranks. Then each engine has the same number of working parts of necessity, and the simple expansive, besides having the usual slide valves, each of which is exposed to the boiler pressure, has an expansion valve to each cylinder, in order to effect so early a cut-off. The following examples will show the results of the two systems under the same circumstances:—

(3) *A simple expansive engine having two cylinders*, each of whose pistons has an area of $\frac{A}{2}$ inches; rate of expansion, 5; initial pressure, 80 lbs.

Mean pressure - - - - = 41·76 lbs.
Effective mean pressure - - = 41·76 − 4 = 37·76 lbs.
Effective initial load on piston = $(80 - 4)\frac{A}{2} = 38\,A$ lbs.
Effective mean load on both pistons = 37·76 × A lbs.
Efficiency of the system - - = 1·00.

(4) *A compound engine having two cylinders*, the ratio of whose piston area is 3, and the area of low-pressure piston, A; rate of expansion, 5; initial pressure, 80 lbs.; pressure in receiver, 21 lbs.

The cut-off in high-pressure cylinder to effect this rate of expansion = $\frac{3}{5}$ or 0·6 stroke.

The cut-off in low-pressure cylinder to maintain 21 lbs. pressure in the receiver = $\frac{80 \times 0{\cdot}6}{21 \times 3}$ = 0·76 the stroke.

Effective mean pressure in H.P. cylinder = 72·48 − 21 = 51·48 lbs.
" " L.P. " = 20·32 − 4 = 16·32 lbs.
Effective initial load on H.P. piston - = $(80 - 21)\frac{A}{3} = 17{\cdot}3 \times A$ lbs.
" " L.P. " = $(21 - 4)\,A = 17{\cdot}0 \times A$ lbs.
Effective mean load on both pistons - = $51{\cdot}48 \times \frac{A}{3} + 16{\cdot}32 \times A$
= 33·48 × A lbs.
Efficiency of the system - - - - = 0·887.

It will be seen that the initial load on each of the pistons of the compound engine is very nearly equal in this case, and is less than half that on each piston of the expansive engine. The work done is very nearly equally divided between the two cylinders, but falls short of that done by the expansive engine by more than 11 per cent.

The compound engine, therefore, is nominally not so economic in steam, but is subject to much lighter loads, and to less variation in load, so that its working parts may be much lighter, and it will run much more steadily than the expansive engine; and owing to such high loads, the friction on the journals and guides will be much more severe with the latter.

There is, however, one other advantage, and that a most important one in actual practice, which the compound engine has over its rival, and that is the comparatively small variation of temperature in the cylinders. The cylinder of the expansive engine is exposed alternately to the heat of the boiler-steam and the chill of the condenser, so that in a very short space of time, in the cases just now considered, the temperature ranges from 312° to 153° Fah., or over 159°.

The high-pressure cylinder of the compound engine dealt with is, however, exposed to a variation of 81° Fah. only, and the low-pressure cylinder to 78° only, or each to only half that of the expansive engine.

The loss of efficiency of the steam from condensation and partial re-evaporation on expansion would be very great in an expansive engine, if it were not steam-jacketed carefully; and with the steam-jacket any re-evaporation which occurs during exhaust is at the expense of the heat from the jackets, which means a loss.

The compound engine, on the other hand, is not *of necessity* fitted with steam-jacketed cylinders, although there is some economy to be effected by doing so. There is, moreover, very great danger of cracking the cylinders with such large variations of temperature, as a complicated structure of cast-iron will not stand sudden cooling when hot, nor much difference between the temperature of one part and the other.

These defects in the expansive engine are magnified and become very serious in slow-moving engines.

For the reasons here already stated that a compound engine is more economic and better working than an expansive engine, the triple compound and quadruple compound engines are superior to the ordinary compound engine.

Relative Consumption of Fuel in the Simple and Compound Engines.—It is claimed, again, for the compound *system*, that economy in consumption of fuel is effected by it. That a great saving may be accomplished by adding a large condensing cylinder to an engine working under the conditions named in example (4), page 77, is undoubted, and needs no further proof; and also that the compound engine using steam of a higher pressure than an expansive engine, and expanding more times, is worked with superior economy, has been shown; but that an expansive engine using the same pressure of steam, and expanding at the same number of times as a compound engine, should be less economical, needs solid proof, inasmuch as theoretically it is seen to be more economical. Great controversies have raged between the supporters of the old engine and those of the compound from the time of Arthur Woolf to that of Dr. Kirk, the former pointing to what theory taught, and the latter to the results of practice; Woolf's engines in Cornwall very early in the last century were showing a "duty" far in excess of Watt's engines, but the philosophers continued to publish books showing that the compound engine was not nearly so efficient. John Elder's compound engines in the Pacific Steam Navigation Co.'s steamers having proved their superiority about 1860, the Admiralty were induced to give the system a trial; they did this, however, in every case at first with steam of 35 lbs. only and engines having more than two cylinders, with plenty of complications. Finally, carefully-devised experiments were made by the British Admiralty, and by the Government of the United States, which show that, although there was no very great difference between the coal consumed per I.H.P. in the two systems, the compound engine, on the whole, is more economical. The best known of these experiments was the trial between the gunboats "Swinger" and "Goshawk," the latter having compound engines, with cylinders 28 inches and 48 inches diameter and 18 inches stroke, the former expansive engines, having two cylinders, 34 inches diameter and 22 inches stroke.

The results of these trials is given in the following table:—

Diameter of cylinder and stroke.	"Swinger."		"Goshawk."	
	Exp., 2 cyls., 34" × 22".		Comp., 28" and 48" × 18".	
Boiler pressure, - - - - - lbs.,	60	49·3	61·6	49·0
Vacuum, - - - - - - ins.,	26·1	26·25	25·5	28·4
Revolutions per minute, - - - -	115	63·53	126	73·3
Mean pressures, - - - - - lbs.,	15·6	...	H.P. 30·7 L.P. 7·65	...
Indicated horse-power, - - - -	363·9	80·4	374	77·8
Pitch of screw, - - - - - -	10′ 2½″	10′ 2½″	9′ 2½″	9′ 2½″
Speed of ship, - - - - knots,	10·21	6·00	10·299	6·00
Coal consumed per I.H.P. per hour, - lbs.,	2·61	2·07	2·603	2·14
Water ,, ,, ,, - -	21·32	21·47	15·57	18·69

These two ships (which were of the same dimensions, form, and power) were tried under similar circumstances, in the full power trial being steamed nearly side by side. Therefore, so far as was possible, the competition was a very fair one, and fully tested the two systems. That the compound engine came out so nearly equal in economic results with the expansive engine is matter for surprise, especially when the above figures are carefully analysed, and really proves the superiority of the compound system, so far as economy of fuel is concerned. The expansive engine had the advantages of a longer stroke and a coarser pitched propeller, which enabled it to obtain both speed and power with fewer revolutions than its rival; and consequently, since the frictional and other losses will be nearly *the same per revolution* in each engine, its loss would be less and its efficiency higher. The compound engine was worked with an unequal distribution of the power, for the ratio of the cylinders being about 3, the work done in the high-pressure is to that done in the low-pressure cylinder as 30·7 to 22·95; this would contribute very much to the losses in it from the excessive "drop" there must have been, and from the high initial load on the high-pressure piston. Although the coal consumed per I.H.P. per hour is practically the same at the full power trial, and somewhat in favour of the expansive engine at the lower power, it was shown, in a paper read before the Institution of Naval Architects by Mr. Sennett, that the consumption of water was much less in the compound engine than in the expansive; and inasmuch as that test eliminates all errors which may have arisen from differences in the boiler, fuel, and stoking, it is a better one. Mr. Sennett also showed that the compound engine could do better still; he gave the results of some further trials made by the Admiralty with gunboats similar to the "Swinger" and "Goshawk," of which the following is a summary :—

Diameter of cylinders and stroke.	"Sheldrake."		"Moorhen."		"Mallard.'	
	Exp., 2 cyls., 34″ × 21″.		Exp., 2 cyls., 34″ × 21″.		Comp. 31″ – 48″ × 18″	
Boiler pressure, - - - - - lbs.,	62·35	53·0	63·5	41	58·4	59·0
Vacuum, . - - - - - ins.,	20	24	23·3	24·4	23·75	25·1
Revolutions per minute, - - - -	115·5	84·7	121·3	92·9	124·8	98·8
Speed of piston, - - feet per minute,	404	295	424	324	374	296
Indicated horse-power, - - - -	367	137	387	180	398	213
Speed of ship, - - . - knots,	9·251	7·232	9·634	7·899	9·894	8·413
Water consumed per I.H.P. per hour, -	21·5	25·4	20·1	24·6	17·12	17·66

Independently, however, of scientific experiments or abstract reasoning, all sea-going ships of the mercantile marine then building were fitted with engines on the compound system; and those few firms who adopted the expansive engine with steam pressure of 60 lbs., abandoned it altogether, and compounded the engines so constructed. The Admiralty, also, soon adopted the compound engine for ships of every size, notwithstanding the argument which was used, with some show of reason then, that the expansive engine was capable of working at full power with steam at atmospheric pressure, which the compound engine could not do, and which it might, for the sake of safety in action, be found advisable, and sometimes necessary, to do. It was considered a very serious thing that ships of the Navy should have engines which could not satisfy this requirement. All scruples on the point, however, were got over by using a three- or four-cylinder compound engine, so arranged that the *whole* of the cylinders could, if so desired, take steam from the boilers direct, and exhaust direct to the condensers.

The compound engine was, after the trials made by the Admiralty, much improved in efficiency, and as more knowledge was gained by experience, all the difficulties with which the early designers were surrounded were overcome, and accepted on all sides, without the least doubt, both as an economical and trustworthy engine. Similar controversies arose on the introduction of the triple-compound engine in 1882, but did not last so long, owing, no doubt, to the better knowledge of the subject in these later days.

Like so many other matters in marine engineering, the questions of the best proportions and arrangements of cylinders are still open, and matters of opinion, and the most successful practice is that in which the best compromise between conflicting conditions has been effected.

Ratio of Cylinder Capacity.—The chief point at issue between rival engineering establishments, as well as between engineers generally, is as to what is the best ratio of cylinder capacity. In designing a horizontal return connecting-rod engine, it has been seen that there is not much choice in the matter, as the power required determines the size of the low-pressure cylinder, and the distance between the piston-rods that of the high-pressure; but when not trammelled by such conditions the ratio may be within certain bounds according to the fancy or judgment of the designer. The low-pressure cylinder is undoubtedly the measure of the power of a compound engine, for so long as the initial steam pressure and rate of expansion are the same, it signifies very little, so far as *total power only* is concerned, whether the ratio between the low- and high-pressure cylinders is 3 or 4 for a compound, or 5 or 8 for a triple; but as the power developed should be nearly equally divided between the cylinders, in order to get a good and steady-working engine, there is a necessity for exercising a considerable amount of discretion in fixing on the ratio.

Whatever be the ratio between the cylinders, it is a comparatively easy matter to set the valves, so that the power shall be equally divided, but in doing this very different results are obtained of another kind, for the initial loads on the pistons will be materially affected. For this reason, in choosing a particular ratio of cylinders, the objects are—to divide the power evenly, and to avoid as much as possible "drop" and high initial loads.

If, in an ordinary compound engine, the power is equally divided, the mean pressure in high-pressure cylinder will be to that in low-pressure cylinder as R, the ratio of their capacities. Neglecting the loss from "drop," the mean pressure in the low-pressure cylinder is always the same, whatever R may be, because for a fixed power the low-pressure cylinder is constant, and half that power is to be developed by it. This being so, as R decreases, the mean pressure in high-pressure cylinder decreases, and that absolutely, not relatively only. But R decreases only by making the high-pressure cylinder larger; and any increase in that direction means increase in initial loads. If the mean pressure is decreased in the high-pressure cylinder with the same initial pressure, the terminal pressure will be less; and since this is so the cut-off in the low-pressure cylinder must be earlier to maintain equal division of the power, which will cause the pressure in the receiver to be raised, producing an increase in the initial load on the low-pressure piston. The increasing of the pressure in the receiver and decreasing of the terminal pressure lessens the "drop," and for this reason some manufacturers make their engines with cylinder ratio of 3 for a boiler pressure of 80 lbs.; while others, to avoid such high initial strains, prefer a ratio of 4.

The "drop" may be reduced to nothing, when the latter is the ratio by "notching up" the links of both cylinders, until the rate of expansion is nearly double that when working full power; the power is then also equally divided, if such is the case at full gear.

If increased economy is to be obtained by increased boiler pressures, the rate of expansion should vary with the initial pressure, so that the final pressure, or that at which the steam enters the condenser, should remain constant. If this rule is adhered to, it follows that, with the ratio of low-pressure to high-pressure cylinder constant, the cut-off in the high-pressure cylinder will vary inversely as the initial pressure, and will be found as follows:—

Let R be the ratio of the cylinders, r the rate of expansion, p_1 the initial pressure.

$$\text{Cut-off in H.P. cylinder} = \frac{R}{r}.$$

r varies with p_1, so that the terminal pressure p_n is constant, and consequently $r = \frac{p_1}{p_n}$.

Therefore, Cut-off in H.P. cylinder $= \text{R} \times \frac{p_x}{p_1}$.

If the cut-off is to remain at about half stroke, it follows that, as $\frac{p_x}{p_1}$ decreases, so R must increase. If no change is made in R, then the cut-off becomes earlier with the increase in boiler pressure, and increased boiler pressure will mean increased strains and increased weights of rods, framing, and shafting.

If, on the other hand, the high-pressure cylinder is decreased as the pressure is increased, so that the initial pressures and mean pressures remain the same on the pistons, there will be the same economy of fuel effected without any material increase in weight of the machinery.

Further Comparison of Efficiency of Engines.—The compound engine, having two low-pressure cylinders and one high, possesses advantages beyond those of the two-cylinder arrangement. It was, no doubt, first chosen principally in order to avoid excessive diameter of low-pressure cylinder for very large power, and because of the uniformity of the twisting moment on the shaft with three cranks at angles of 120° apart. But in this engine the work can be equally divided between the cylinders without those disadvantages previously shown to exist in the two-cylinder engine working under this condition. To divide the work equally, only one-third will be allotted to the high-pressure cylinder, and one-third to each of the two low-pressure cylinders, and this is done by maintaining a considerably higher pressure in the receiver than obtains in the two-cylinder arrangement. The "drop," therefore, is considerably less; and since each low-pressure piston has only half the area of that of the two-cylinder engine, the initial loads are not abnormally large. The increase in receiver pressure reduces the initial load on the high-pressure piston, and hence its diameter may be increased, so as to get increased expansion in it without increasing the initial load beyond that on the high-pressure piston of a two-cylinder engine of equal power.

(5) *An engine having two low-pressure and one high-pressure cylinders working on three cranks.*—To fully appreciate these advantages, suppose such a three-cylinder engine to be working under the same circumstances as that of example (4), page 83, so that the area of each low-pressure piston is $\frac{\text{A}}{2}$, and of the high-pressure piston $\frac{\text{A}}{3}$, the rate of expansion 5, and the initial pressure 80 lbs., absolute.

As in the previous example the cut-off is 0·6 the stroke. Suppose the pressure in the receiver to be 32 lbs. absolute,

Then the cut-off in L.P. cylinders $= \frac{80 \times 0{\cdot}6}{32 \times 3} = 0{\cdot}56$ the stroke.

The effective mean pressure in the H.P. cylinder - - - - - - - $\Big\} = 72{\cdot}48 - 32 = 40{\cdot}48$ lbs.

The effective mean pressure in each L.P. cylinder - - - - - - - $\Big\} = 28{\cdot}19 - 4 = 24{\cdot}19$ lbs.

Effective initial load on the H.P. piston $= (80 - 32)\frac{\text{A}}{3} = 16 \times \text{A}$ lbs.

" " each L.P. " $= (32 - 4)\frac{\text{A}}{2} = 14 \times \text{A}$ lbs

The effective mean load on all three pistons - - - - - - $= 40{\cdot}48 \times \frac{A}{3} + 2 \times 24{\cdot}19 \times \frac{A}{2}$ $= 37{\cdot}68 \times A$ lbs.

Efficiency of the system - - - - - = 0·998.

It is seen by this that the initial load on the high-pressure piston is 7½ per cent., and that on each low-pressure piston 17½ per cent., less than the corresponding loads of the two-cylinder engine of the same size; that the gain of efficiency of the steam is 12½ per cent., if no allowance is made for superheating of the steam on expanding into the receiver; the "drop" in this case is 54 – 32, or 22 lbs., as against 33 lbs. in the two-cylinder engine. It is seen, then, that theoretically this engine is nearly equal in steam efficiency, both to the simple expansive, and to the compound direct-expansion engine. On the other hand, the friction of three engines must be set against this, besides the extra cost of manufacture and the space occupied by machinery.

Triple-Expansion Compound Engine.—The compound engine having one high-pressure, one low-pressure, and one medium-pressure cylinder, is the one most commonly found in use; and if steam of over 100 lbs. pressure is to be used economically, and if the engine is to be efficient and good-working, some such arrangement is necessary. To ensure success, the steam must be expanded down to about 10 lbs. absolute, the initial loads on the pistons moderate, and the drop not excessive. The low-pressure cylinder will be rather smaller in size than that of the ordinary *two*-cylinder arrangement (the reduction depending on the increased efficiency of the steam from the large rate of expansion).

(6) For example:—To determine the particulars of a three-cylinder compound engine on this system to be equal to that set out in example (4), page 83, the initial pressure being 127 lbs., and the rate of expansion 10.

The mean pressure in a single cylinder, with a cut-off at $\frac{1}{10}$ the stroke, is 41·91 lbs.; deducting from this 4 lbs. for back pressure, the mean *effective* pressure is 37·91 lbs., or nearly the same as that of example (5), page 88. Suppose the cut-off in the high-pressure cylinder is 0·6 the stroke, then the ratio of the high-pressure to low-pressure cylinder must be 6 to effect a rate of an expansion of 10. The ratio of the medium-pressure cylinder to high-pressure cylinder may be taken as $\frac{5}{2}$, and, consequently, the ratio of low-pressure to mean-pressure cylinder is $\frac{12}{5}$. The pressure between the high-pressure and mean-pressure cylinders is to be 50 lbs., the cut-off in the mean-pressure cylinder will therefore be 0·61 the stroke. The pressure between the low-pressure and mean-pressure cylinders is to be 21 lbs.; the cut-off in low-pressure cylinder must therefore be 0·6 the stroke.

The effective mean pressure in H.P. cylinder = 115 – 50 = 65 lbs.
" " M.P. " = 45·4 – 21 = 24·4 lbs.
" " L.P. " = 19 – 4 = 15·0 lbs.

The effective initial load on H.P. piston $= (127 - 50)\frac{A}{6} = 12{\cdot}83$ A lbs.
" " M.P. " $= (50 - 21)\,A\,\frac{5}{12} = 12{\cdot}1$ A lbs.
" " L.P. " $= (21 - 4)\,A = 17{\cdot}0$ A lbs.

The effective mean load on all three pistons $= 65 \times \frac{A}{6} + 24{\cdot}4 \times \frac{5}{12} A + 15\,A = 36 \times A$ lbs.

Efficiency of the system = 0·949.

It will be seen that in this case, owing to "drop," there is a loss of nearly 2 lbs., but the work is fairly divided between the three cylinders, and the initial loads are by no means high; the drop from the high-pressure cylinder is only 26 lbs., and that from the mean-pressure cylinder 9½ lbs. This engine then effects an expansion of 10, and is therefore a very economic one; it will have a very regular motion, and share in all the benefits of a three-cylinder engine; and the strains on the working parts will be very moderate.

(7) To see how far this is true, it is only necessary to compare the results with those from *a three-cylinder engine having two low-pressure cylinders*, each having a piston area of $\frac{A}{2}$.

Suppose the high-pressure cylinder to have a piston-area of $\frac{A}{4}$, the cut-off in the high-pressure cylinder to effect an expansion of 10, must be 0·4 of the stroke; the receiver pressure will be 42 lbs., and the cut-off in low-pressure cylinder 0·3 of the stroke.

The effective mean pressure in the H.P. cylinder - } $= 97 - 42 = 55$ lbs.

The effective mean pressure in each L.P. cylinder - } $= 27{\cdot}7 - 4 = 23{\cdot}7$ lbs.

The effective initial load on the H.P. piston - - } $= (127 - 42)\frac{A}{4} = 21{\cdot}25 \times A$ lbs.

The effective initial load on each L.P. piston - - } $= (42 - 4)\frac{A}{2} = 19{\cdot}0 \times A$ lbs.

The effective mean load on all three pistons - - } $= 55 \times \frac{A}{4} + 2\left(23{\cdot}7 \times \frac{A}{2}\right) = 37{\cdot}45 \times A$ lbs.

The initial load on the high-pressure piston is here 65 per cent. larger than in the preceding case, and that on each low-pressure piston 56 per cent. larger than on the mean-pressure piston, and 11¾ per cent. above that on the low-pressure piston; but the efficiency of the steam is somewhat higher in the latter case, the drop from the high-pressure cylinder being only 8·8 lbs. The ratio of maximum pressure to mean in the high-pressure cylinder is 1·54, and in the low-pressure cylinder 1·6, which are about the same as those of the old expansive engine working with a boiler pressure of 45 lbs., and cutting-off at 0·3 of the stroke. It is not always advisable to set the cranks at angles of 120° in a three-crank compound engine; their precise position depends on the power developed in each cylinder, and the relative twisting efforts at any moment.

The success of the triple-expansion engine is now so well assured, and all doubts as to its efficiency and good working are so effectually dispelled, that it is without doubt the engine of the day. It does not differ in any essential feature from the ordinary compound engine, and its success was in no small measure assured by the fact that most makers of the new type departed as little as possible from their previous practice in its general construction. The arguments for and against this type of engine bear a striking resemblance to those used in the well-remembered warfare of Compound *versus* Expansive engines, and the objections most strongly insisted on by the opponents of this system are just those used against the original compound engine, and are rather the echo of old battle-cries than the sounds of new ones. A few years' experience demonstrated that the triple-expansion engine is more economical than the ordinary compound engine; that the wear and tear is no more, but rather less, when three cranks are employed, than with the two of the ordinary compound; and that boilers of the common marine design could be made to work satisfactorily at a pressure of 150 lbs. per square inch, and even higher, while, with ordinary care, their durability and good continued working are not likely to be less than those of similar boilers pressed to 60 lbs. per square inch under similar circumstances. Speaking generally, the consumption of fuel is 25 per cent. less with a triple-expansion engine than with an ordinary compound engine working under similar circumstances. That is, a triple-expansion engine, supplied with steam at 150 lbs. pressure, uses 25 per cent. less weight of water per I.H.P. than an ordinary compound engine supplied with steam at, say, 90 lbs. pressure, both engines being equally well-designed, manufactured, and attended to. Also, that a triple-expansion engine is more economical than an ordinary compound engine when both are supplied with steam at the same pressure, for all pressures of 95 lbs. and upwards, and especially so in the case of large engines. Hence, it may be taken that the superior economy of the triple-expansion engine, as now constructed, is due to two causes, viz.:—(1) To the superior pressure of steam used and the higher rate of expansion thereby possible; and (2) the system whereby large initial loads and large variations of temperature in the cylinders and large "drop" in the receivers are avoided.

Increased pressure of steam is obtained by a very slight increase of consumption of fuel, and the efficiency of steam rapidly increases with increased pressure; hence, steam of high pressure is more economical than that at inferior pressures. For example:—

(i.) The total heat of evaporation of 1 lb. from 100° and at 274° F. (corresponding to 45 lbs. pressure absolute) is 1097 thermal units.

(ii.) From 100° and at 320° F. (corresponding to 90 lbs. absolute) is 1110 thermal units.

(iii.) From 100° and at 353° F. (corresponding to 140 lbs. absolute) is 1120 thermal units.

(iv.) From 100° and at 377° F. (corresponding to 190 lbs. absolute) is 1127 thermal units.

Suppose in each case the steam to be expanded to a terminal pressure of 10 lbs. absolute, the rates of expansion will then be 4·5, 9, 14, and 19 respectively; and the mean pressures corresponding to these initial pressures and rates of expansion will be 25 lbs., 32 lbs., 36 lbs., and 39 lbs. respectively. If the volume of a pound of steam varied exactly in the inverse ratio of the pressure, these figures would represent the relative values of the efficiency of the steam at the various pressures. But, taken exactly, the relative values are 25, 33·3, 38·5, and 42·6, thus showing that a pound of steam at 90 lbs. pressure is capable of doing 33 per cent. more work than a pound at 45 lbs.; a pound of steam at 140 lbs. pressure, 16 per cent. more than a pound at 90 lbs.; and a pound at 190 lbs. pressure, 10·6 per cent. more than a pound at 140 lbs. pressure, or 28 per cent. more than at 90 lbs. pressure. In other words, an engine using steam at 140 lbs. pressure should, apart from any practical considerations, consume 16 per cent. less fuel than one using steam at 90 lbs.; and, again, that an engine using steam at 190 lbs. should consume 28 per cent. less fuel than one using steam at 90 lbs., and 10·6 per cent. less fuel than one using steam at 140 lbs.

Looking to see how far practice agrees with these results, it is found that the ordinary compound engine, using steam at 140 lbs., is little, if at all, more economical than one using steam at 90 lbs., while the triple-expansion engine, with steam at 140 lbs. pressure, gives a greater economy than theory shows should be due to the increased pressure. It follows, then, that there is some other cause operating to produce the economic results shown in every-day practice with this new engine, for there is now no question that the saving in fuel effected by a triple-expansion engine, using steam and expanding 11 or 12 times, is about 25 per cent. of that used by an ordinary compound engine of the same power, using steam at 90 lbs. and expanding 8 to 9 times. The other cause, or rather causes, are not far to seek, for it is to be noticed at once that since, by using steam in the two cylinders of a compound engine, the large variation in temperature in the cylinder of the expansive engine was avoided (and this, doubtless, was one of the chief causes of its superior economy over the latter), then, by using three cylinders for the higher pressures, a similar result would be obtained. Further, as the ordinary compound engine is not subject to such severe initial and working strains as prevail in expansive engines of the same power, and using steam of the same pressure, so, in the triple-expansion engine, with three cranks, these strains are still further reduced. In other words, by extending those leading features of the compound engine which conduced to its economy, the engineers have achieved, with the triple expansion, a victory similar to that obtained thirty years ago by their predecessors, but with somewhat less brilliant results; and it is not difficult to see that any further advances must meet with still less gain. Until science and skill have discovered new materials, or other applications of old ones, there will not be much practical advantage in using steam at higher pressures than now obtained, and 200 lbs. pressure seems about the limit at which further theoretical economy is swallowed up by practical losses.

It has been repeatedly maintained by the opponents of triple-expansion engines that, if the ordinary compound engine is designed with a long stroke, it is as economical. In order to see how far this is true by practice, it is sufficient to examine the diagrams of the engines of the s.s. "Northern," whose cylinders were 26 inches and 56 inches diameter and 60-inch stroke, used steam at 130 lbs. absolute, and indicated 1235 H.P., which show a consumption of 15·4 lbs. of water per I.H.P. per hour; and those of s.s. "Draco," whose engines have cylinders 21 inches, 32 inches, and 56 inches diameter and 36-inch stroke, using steam at 125 lbs. absolute, and indicating 618 H.P., which show a consumption of 14·1 lbs. per I.H.P. per hour. Or, again, by comparing the performance of the "Draco" with that of the "Kovno," having engines whose cylinders are 25 inches and 50 inches diameter and 45-inch stroke, using steam at 105 lbs. pressure absolute, and indicating 809 H.P., which show a consumption of 16·6 lbs. of water per I.H.P. In these cases, the consumption of the "Kovno" is 17·73 per cent. in excess of that of the "Draco," and 7·58 per cent. in excess of that of the "Northern"; and the "Northern" consumes 9·4 per cent. more than the "Draco."

Comparing two other engines of smaller size than the above, but indicating nearly the same power, the gain is still more marked. The "Dynamo" with triple-expansion engines, whose cylinders are 17 inches, 27 inches, and 48 inches diameter and 27-inch stroke, supplied with steam at 165 lbs. absolute pressure, and indicated 638 H.P. The "Bolama" had ordinary compound engines, whose cylinders were 24 inches and 45 inches diameter and 33-inch stroke, supplied with steam at 95 lbs. absolute pressure, and indicated 561

H.P. The consumption of water of the "Bolama" was 16·8 lbs. per I.H.P., against 13·0 lbs. per I.H.P. of the "Dynamo," or 29·2 per cent. in favour of the triple-expansion engine.

It is, however, needless now to multiply cases, as it is a matter of common observation that the saving in fuel is from 20 to 30 per cent., and it may safely be taken that the triple-expansion engine, using steam at 165 lbs. absolute pressure, uses 25 per cent. less fuel than an equally good ordinary compound engine of equal power and working under similar circumstances, using steam at 100 lbs. absolute pressure. That the variation in temperature in each cylinder of a triple-expansion engine must be less than in each cylinder of an ordinary compound engine with the same boiler pressure, is manifest, inasmuch as there is the same difference between the initial and terminal temperatures, and in the one case this is divided into three, and only into two in the other. The same, however, holds good when the boiler pressure of the triple is 165 lbs. absolute, and that of the ordinary compound is only 100 lbs.; for in the former case the total variation is 240° F., which divided by 3 gives a variation in each cylinder of 80° F., and in the latter case it is 202° F., which divided by 2 gives a variation in each cylinder of 101° F., or 26 per cent. excess.

On examining the indicator diagrams of the "Northern," it is found that the variation of temperature in the high-pressure cylinder is 102°, and in the low-pressure 100°; while the diagrams of the "Draco" show a variation of 64° in the high-, 68° in the medium-, and 66° in the low-pressure cylinder. Those of the "Kovno" show a variation of 93° in the high-, and 100° in the low-pressure cylinder.

As the three-crank triple-expansion engine is now accepted as the most suitable for marine practice generally, it will be better to compare it, so far as practical considerations are concerned, with the ordinary compound, and for that purpose it is assumed in each case that the triple is supplied with steam of 150 lbs. pressure by the gauge, or 165 lbs. absolute, unless stated to the contrary. Suppose, now, two engines are to be taken for comparison—viz., a triple-expansion and an ordinary compound—to develop equal powers with the same stroke of piston and same diameter of low-pressure cylinder. The initial pressure in the one case is to be 100 lbs. absolute, and in the other 165 lbs. absolute. Let the number 14 represent the area of the low-pressure piston in each case; the mean pressure referred to the low-pressure piston be 24, and the efficiency of the systems equal, so far as expanding the steam is concerned. The area of the high-pressure piston may then, without fear of controversy, be taken as 4 for the ordinary compound, and 2 for the triple, and the area of the medium-pressure piston of the triple as 5. If the referred mean pressure is equally divided in each case between the cylinders, the following shows the relative work done, viz.:—

ORDINARY COMPOUND ENGINE.		TRIPLE-EXPANSION ENGINE.	
High-pressure cylinder,	4 × 42 or 168	High-pressure cylinder,	2 × 56 or 112
Low ,,	14 × 12 or 168	Medium ,,	5 × 22·4 or 112
		Low ,,	14 × 8 or 112

That is, the *average* stress on the rods, columns, guides, &c., is 50 per cent. more with the ordinary compound engine than with triple-expansion.

To obtain a mean pressure of 24 lbs., with an initial pressure of 165 lbs. absolute, and a pressure in the condenser of 2 lbs., the rate of expansion is 14, with an efficiency of 0·6; and with an initial pressure of 100 lbs. the rate of expansion is 7.

On examining diagrams taken under these circumstances from actual engines, the following is to be observed:—

Initial pressure in the high-pressure cylinder of the compound engine, 98 lbs.; back pressure, 23 lbs.; effective initial pressure, 75 lbs. per square inch; or load on the piston, 75 × 4, or 300. In the low pressure the initial pressure is 22 lbs.; back pressure, 4; giving an effective initial pressure of 18 lbs. per square inch; or load on the piston, 18 × 14, or 252.

The initial pressure on the high-pressure cylinder of the triple-expansion engine is 160 lbs.; back pressure, 63 lbs.; effective initial pressure, 97 lbs.; or the load on the piston, 97 × 2, or 194. In the medium pressure the initial pressure is 70 lbs.; and the back pressure, 21 lbs.; effective initial pressure, 49 lbs.; or the load on the piston, 49 × 5, or 245. In the low pressure the initial pressure is 18 lbs.; and the back pressure 4 lbs.; giving an effective initial pressure of 14 lbs. per square inch; or a load on the piston of 14 × 14, or 196. Thus showing the loads in the case of the triple-expansion engine to be much less, notwithstanding the higher pressure of steam employed.

This, too, is shown in actual practice by comparing the initial loads on the engine of the s.s. "Northern," whose cylinders are 26 inches and 56 inches diameter and 60-inch stroke, indicating 1242 H.P., and supplied with steam at 130 lbs. pressure absolute, with those of the triple-expansion engine of the s.s. "Ariel," whose cylinders are 23 inches, 35 inches, and 60 inches diameter and 57-inch stroke, indicating 1527 H.P., and supplied with steam at 165 lbs. pressure absolute.

The "Northern's" high-pressure piston sustains an initial load of 530 × 100, or 53,000 lbs.; the low-pressure, 2463 × 24, or 59,112 lbs. The "Ariel's" high-pressure piston sustains 415 × 100, or 41,500 lbs.; the medium-pressure, 962 × 60, or 57,720 lbs.; and the low-pressure, 2827 × 18, or 50,886 lbs.—notwithstanding that the engines are larger and develop nearly 25 per cent. more power with higher boiler pressure.

The more even distribution of pressure also very materially affects the resistance of the slide-valves, and so tends in every way to reduce the losses due to mechanical causes.

So far, experience has shown that the wear and tear of the triple-expansion engine with three cranks is very considerably less than with the ordinary two-crank compound engine of the same power and stroke, and no doubt this is due to those causes already shown to exist with this class of engine.

The three-crank triple-expansion engine has, however, shown another valuable quality, and one which may easily be surmised from the foregoing reasoning—viz., that a much higher indicated power may be developed with a low-pressure cylinder, equal in size to that of a common compound engine, without any increase in the initial loads on the pistons. This may be shown by comparing the performance of the engines of the "Eldorado," whose cylinders are 26 inches, 40 inches, and 68 inches diameter and 39-inch stroke, supplied with steam at 165 lbs. absolute pressure, with that of the engines of the "Juno," whose cylinders are 35 inches and 69 inches diameter and 39-inch stroke, supplied with steam at 100 lbs. absolute pressure. When running at 72 revolutions, the "Eldorado's" engines develop 1572 I.H.P., and the "Juno's" 1249 I.H.P., or 26 per cent. more power from the triple-expansion than from the ordinary compound, and that with a low-pressure cylinder 3 per cent. smaller. The initial loads on the pistons of the "Eldorado" are

54,060 lbs., 66,568 lbs., and 61,727 lbs., as against 69,264 lbs, and 71,041 lbs. on those of the "Juno."

Similar results can be shown with other engines of various sizes; and to extend the question, it may be taken as approximately correct that a referred mean pressure may be used in a triple-expansion engine 50 per cent. higher than with an ordinary compound engine without any serious difference in the stresses on the working parts and frame-work. It is for this reason possible to manufacture a triple-expansion engine at the same price per I.H.P. as an ordinary compound engine; and, also, since the efficiency of a three-crank engine is higher than that of a two-crank engine, the economy is higher than that due to the saving in fuel per I.H.P. alone. The propelling efficiency of the three-crank engine is especially noticeable when running at low speeds, and it is no doubt on this account the best for naval purposes where so much cruising is done at comparatively slow speeds. It is also capable of being worked at much fewer revolutions without stopping on the centres than a two-crank engine, which is highly advantageous in navigating intricate channels and docks and during a fog, as steerage power is kept without much "way" on the ship. This engine is also, when well constructed and properly adjusted, almost noiseless, and causes little or no vibration, which is an advantage in every ship, but especially in yachts and passenger steamers.

Mean Pressure in a Triple-Expansion Engine is from 0·6 to 0·7 of that given by theoretical calculation, and, as may be supposed, depends very much on the port areas, valve openings, and such other conditions as affect ordinary engines. At the high rates of expansion usually adopted, when economy of fuel is of first importance, there is a very strong tendency to condensation in the cylinders, and this is especially noticeable when running at low speeds. It seems from present experience that the more quickly the steam is caused to pass through the system of cylinders, the higher its efficiency. From calculations made from indicator diagrams, it is seen that the condensation, or rather partial condensation, from the high-pressure to the low-pressure cylinder amounts to 3·2 lbs. per I.H.P. in the "Rosario" at 62 revolutions, is 2·3 lbs. in the "Dynamo" at 90 revolutions, and only 0·6 lb. in the "Mosquito" at 102 revolutions; again, in the "Finland" at 72 revolutions the loss is 2·3 lbs., whilst, when notched up and running at 67 revolutions, it is 4·29 lbs., and in the "Electro" at 89 revolutions it is 2·63 lbs., but at 72·5 revolutions it is as much as 5·8 lbs.; whilst in the ordinary compound engines of the "Bolama," it is only 1·5 lbs. at 82 revolutions, and 2·1 lbs. at 66·5 revolutions. Had the cylinders been steam-jacketed, the loss would no doubt have been less; and, therefore, all slow-running triple-expansion engines should have at least the medium-pressure cylinder steam-jacketed, or use superheated steam, while fast-running engines can do very well without it.

If the steam ports are such that the flow at exhaust does not exceed 6000 feet per minute for moderate sizes of cylinders, and 5000 for small sizes, and the opening to steam such that the flow is not more than 50 per cent. in excess of these, and the engines are to be run at a fairly high speed, the actual mean pressure should be 0·66 of the theoretical, and when the port area is more than given by the above rules, and a high number of revolutions, it should be 0·7 of the theoretical. The ordinary mercantile marine engine, as generally designed, gives a coefficient of 0·62 to 0·66, while some are as low as 0·6.

Example.—To find the mean pressures expected from a triple-expansion engine having good ports, and to be worked at a fairly high number of revolutions; the boiler pressure is 160 lbs. absolute, and the pressure in the condenser 2 lbs., the total rate of expansion being 12.

Referring to Table ix., page 109, the coefficient for an expansion of 12 is 0·2904; hence,

Theoretical mean pressure is $174 \times 0{\cdot}2904 - 2$ or 48·5 lbs.
The actual mean pressure is $48{\cdot}5 \times 0{\cdot}66$ or 32 lbs.

Example.—To find the size of cylinders to indicate 1200 horse-power under the above conditions with a piston speed of 600 feet per minute.

$$\text{Area of low-pressure piston} = \frac{1200 \times 33{,}000}{32 \times 600} = 2062 \text{ square inches.}$$

And the diameter is $51\frac{1}{4}$ inches.

The power is supposed to be equally divided between the three cylinders, hence the mean pressure in the low pressure is 10·67 lbs.

If this rate of expansion is to be obtained with a cut-off at 0·6 of the stroke in the high-pressure cylinder, then the area of high-pressure piston

$$= \frac{2062}{0{\cdot}6 \times 12} = 286 \text{ square inches, and the diameter } 19\tfrac{1}{8} \text{ inches.}$$

The ratio of the low- to high-pressure cylinder is here 7·2, and, therefore, the mean pressure in the high-pressure cylinder is $7{\cdot}2 \times 10{\cdot}67$, or 76·8 lbs.

The size of the medium-pressure cylinder is a subject on which most engineers seem to differ. The mean size is, of course, obtained by dividing the area of the low-pressure cylinder by the square root of the ratio between the low and high. So that in this case the area of the medium-pressure piston

$$= \frac{2062}{\sqrt{7{\cdot}2}} = 769 \text{ square inches, and diameter } 31\tfrac{1}{4} \text{ inches.}$$

From practical considerations this is too large, as the ranges of temperature will be unequal, and the loads unequal also, with such proportions; so that in the case here taken, 30 inches would be quite large enough.

Rules for Sizes of Cylinders.—The area of the high-pressure piston

$$= \frac{\text{area of low-pressure piston}}{\text{cut-off in H.P. cylinder} \times \text{rate of expansion}}.$$

Area of the medium-pressure piston

$$= \frac{\text{area of low-pressure piston}}{1{\cdot}1 \times \sqrt{\text{ratio of L.P. to H.P. cylinder}}}.$$

In general practice the ratio of low-pressure to high-pressure depends on the pressure of steam supplied and the service of the engine. In the ordinary mercantile steamer where economy in fuel is of first consideration, and the boiler pressure is 150 to 165 lbs., the ratio is 6 to 6·8; when the pressure is 170 to 200 lbs., the ratio is 7 to 8·0. In the case of fast cross-channel steamers, warships, and others, where power from a limited weight of machinery is of great importance, the ratios will be 4·5 to 5·6 when the steam pressure is 150 to 165, and 6 to 7·0 for the higher pressures up to 200 lbs. In fact, in warships using steam from 220 to 250 lbs. pressure, the ratio is usually 7. In such ships the ratio of low-pressure to medium-pressure cylinders is 2·7. In the mercantile marine the ratio is often about the square root of the ratio of low-pressure to high-pressure, but this, as already stated, gives a cylinder with too high initial loads; a better size is given by making the ratio = $1{\cdot}1 \times \sqrt{(\text{ratio of L.P. to H.P.})}$. In quadruple engines the ratio of low-pressure to high-pressure cylinder is usually 8; the diameter of first medium-pressure is half that of low-pressure cylinder, and the diameter of second medium-pressure is twice that of the high-pressure cylinder.

The space occupied by the triple-expansion engine was one of the special and valid objections raised against it by its detractors, but when closely looked into, it is not found to be by any means a vital hindrance. The objection cannot, of course, be raised at all against this engine when used with two cranks, but only when three are employed.

It requires little or no argument to make plain what is at first sight obvious—viz., that three cranks must take up more space in the direction of the axis of the shaft than do two; but seeing how often in a cargo boat the engine-room is made unnecessarily large to obtain the 32 per cent. reduction in tonnage, the objection in this case is without weight. Even when this is not the case, and the engine-room has to be prolonged to suit the engine, the arrangement also gives space for larger engine-room bunkers, and thus permits of reduction of the athwart bunker, if it does not allow of its total abolition. Moreover, since the reduction in consumption of fuel is from 20 to 30 per cent., to provide for the same number of days' steaming, the bunkers may be from 20 to 30 per cent. smaller; or, with the same amount of bunker space, the ship can steam from 20 to 30 per cent. longer, which is by no means the least of the advantages claimed for the triple-expansion engine, and a most valuable one in the case of ships making long voyages. This consideration will, doubtless, materially affect the prosperity of many of the foreign coaling stations, to the material advantage of the shipowner and, to some extent, a blessing to the passenger.

Again, as a set-off to an enlargement of the engine, the saving in consumption of fuel permits of a corresponding reduction in the size of, and consequently the space occupied by, the boilers.

And, finally, since it is possible that as much as 50 per cent. more power can be developed with the triple-expansion engine than with the ordinary compound having the same size of low-pressure cylinder; and, as very generally from 20 to 25 per cent. more power is developed for equal nominal horse-power, an engine about 20 per cent. smaller has been generally adopted. This is easily understood when it is known that, with a good triple-expansion engine, a referred mean pressure of 33 to 40 lbs. can be obtained as against 25 lbs. under similar circumstances with an ordinary compound engine, and 30 lbs. against 22 lbs. with the most ordinary description of engines.

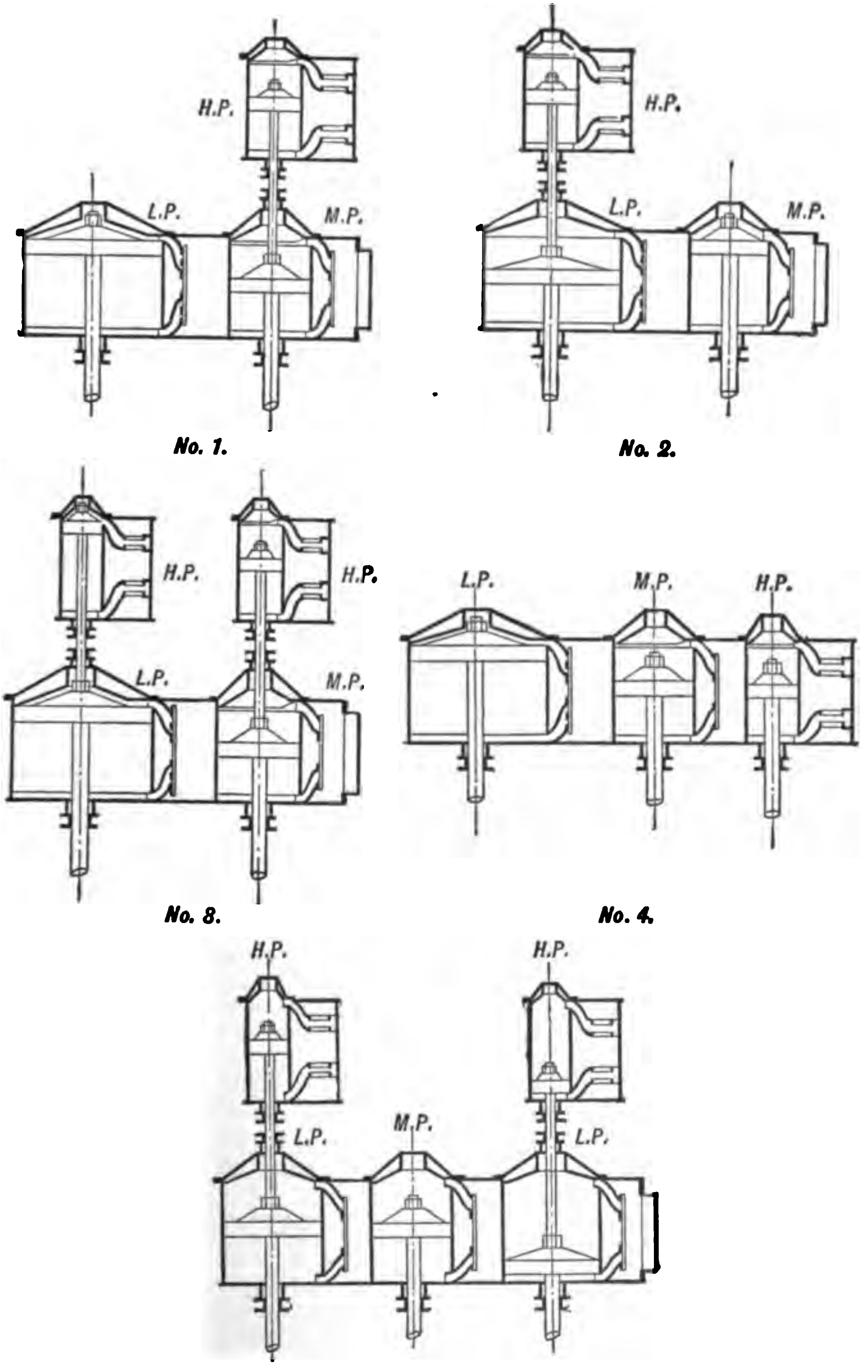
To meet objectors, some engineers went so far with alteration of design as to run a very great risk of bringing the triple-expansion engine into bad repute. Beyond doubt it is of much more importance to the shipowner to have an engine which works with little wear and tear, and without breakdown and damage, than to secure a few extra feet of hold space. It is, therefore, much to be deprecated that the crank-pins and journals of these engines should be so shortened as to require constant attention, their brasses being almost inaccessible. It is true that this type of engine permits of smaller crank-pins, owing to the comparatively small initial loads, while the journals for the same reason (and because of the superior balance of the power) may also be shorter, and it is equally true that it is folly to adhere blindly to rules which only apply to the two-crank engine when designing a three-crank engine; nevertheless, on the other hand, it must not be overlooked that by following rules based only on working considerations and proportioned to normal stresses, an engine may be designed which is so crowded and confined as to render that attention to the working parts which is so necessary at all times almost an impossibility. In other words, an engine of this sort may be theoretically, or rather scientifically, perfect, but practically bad.

Experience has shown that a triple-expansion engine may be designed with ample bearing surfaces, easy of access, and even with valves worked by link motion, and still occupy but very little more space than an ordinary compound engine of the same I.H.P., and that some of these engines, really taking up no more space than the old type, give most satisfactory results.

Arrangement of Cylinders.—The different ways in which the cylinders of a triple-expansion engine may be arranged are very numerous. The first engine of the kind, designed by Mr. Kirk for the s.s. "Propontis," had cylinders 23 inches, 41 inches, and 62 inches diameter by 42-inch stroke, working on three cranks. These engines were made in 1874, but owing to the failure of the boilers were set down generally as unsuccessful. It was not till advances in the manufacture of steel and a better knowledge of boiler-making had enabled engineers to construct boilers of the ordinary cylindrical tubular type for much higher pressures than had hitherto generally obtained, that Messrs. Douglas & Grant could venture again, in 1878, to take up the triple-expansion engines for the yacht "Isa." These engines have cylinders 10 inches, 16 inches, and 28 inches diameter by 24-inch stroke, supplied with steam at 120 lbs. pressure, from a cylindrical boiler 8 feet 9 inches diameter, and 8 feet 6 inches long; the cylinders were, in this case, arranged with the high-pressure above the medium-pressure operating on one crank, and the low-pressure operating on the other. In the year 1881 Mr. Kirk again designed a three-crank triple-expansion engine for the s.s. "Aberdeen," having cylinders 30 inches, 45 inches, and 70 inches diameter by 54-inch stroke; the steam pressure was, however, only 115 lbs. per square inch.

Taking the two-crank engine, two methods are of course very obvious, and are shown by diagrams Nos. 1 and 2 on p. 99. No. 1 is the arrangement of the s.s. "Isa," and likewise that of the "Draco" previously spoken of. It is a simple plan, and, of course, has the merit of taking up only the same space as the ordinary compound engine. The method is, however, open to the objection that it is almost impossible to divide equally the power between the two cranks, and, when the power is only divided as 2 to 3, the initial loads are comparatively high. This and the method shown in diagram No. 2 are now chiefly noticeable as a cheap and convenient way of recompounding old compound engines, and adapting them to work with steam of higher pressures. That the power may be evenly divided between two cranks, two high-pressure cylinders may be employed, one over the medium-, and one over the low-pressure cylinder, as shown on diagram No. 3. This, too, is a convenient way of recompounding old engines, especially when of large power, where the disparity of loads would be serious if plans No. 1 or 2 were adopted.

If it is desired to have only two cranks, and the engines are of power so large as to require an abnormally large low-pressure cylinder, then plan No. 1, p. 100, may be adopted with advantage, for by it there are two low-



No. 1.

No. 2.

No. 3.

No. 4.

No. 5.

Fig. 22.—Various Arrangements of the Cylinders of Triple-Expansion Engines.

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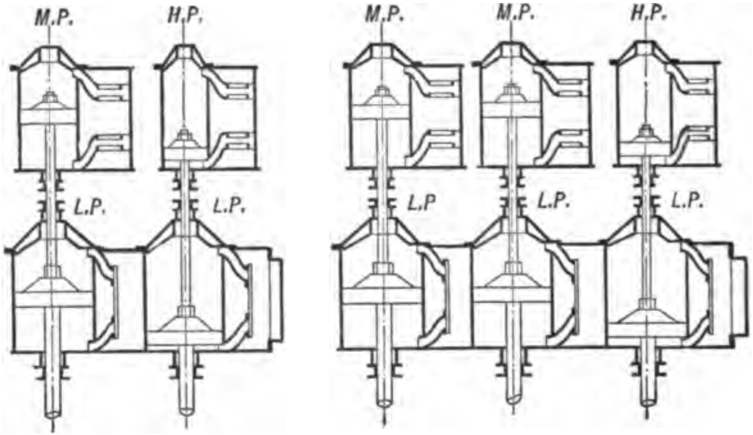
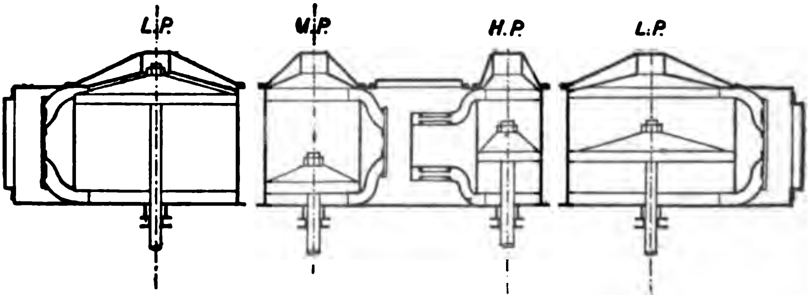
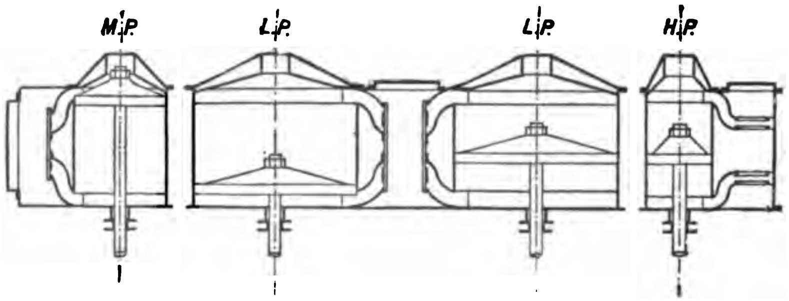


No. 1. No. 2.



No. 3.
Four cranks, L.P. rods reduced.



No. 4.
Four cranks, all rods same size.

Fig. 22*a*.—Various Arrangements of the Cylinders of Triple-Expansion Engines.

pressure cylinders, with the high-pressure cylinder over one, and the medium-over the other. These two latter plans, however, have the disadvantage of four cylinders as against three, with the corresponding number of valves and stuffing-boxes to absorb power.

Plan No. 4, p. 99, shows the arrangement of cylinders in their natural sequence for a three-crank engine, and is now most generally adopted. If the valve-boxes are placed at the front or back of the engines, the cylinders may be much closer together, and a very symmetrical arrangement is got by placing the low-pressure cylinder in the middle, while a little less space still is occupied by putting the high-pressure cylinder in the middle.

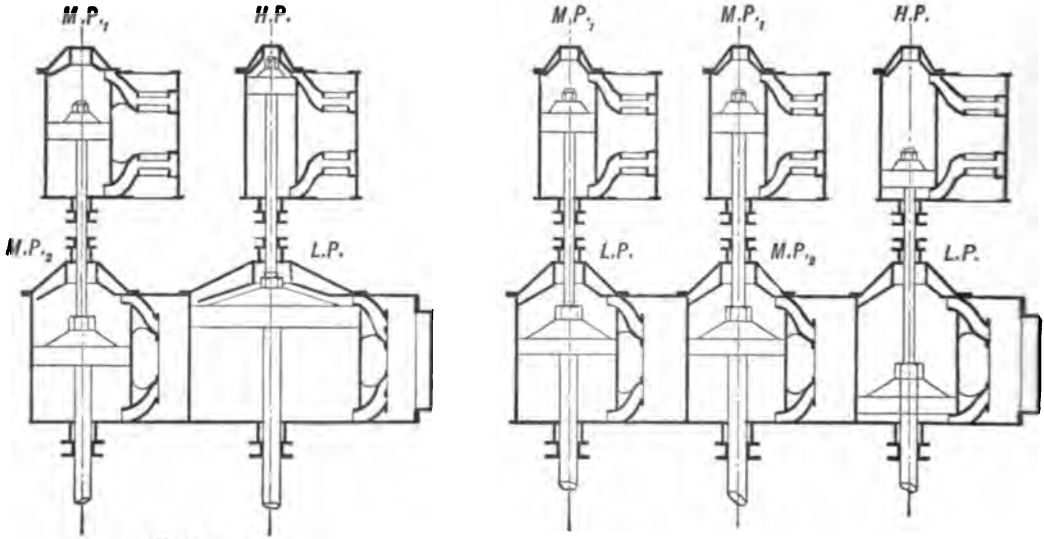
Plan No. 2, p. 100, is an arrangement suitable for very large engines when it is necessary to have three low-pressure cylinders—in this case over each there is a cylinder, two of which are medium-pressures, and one a high-pressure, and for simplicity these three might be of one size.

Plan No. 5, on p. 99, shows another arrangement suitable for engines of large size, and also by which large engines, having three cranks and two low-pressure cylinders may be recompounded, and still have the power equally divided between the cranks. This is done by placing a new high-pressure cylinder over each of the two low-pressure cylinders. Another obvious method is to place one new high-pressure cylinder over the original high-pressure, which becomes thus converted as before into a medium-pressure cylinder; in this case, however, the power would be unevenly divided.

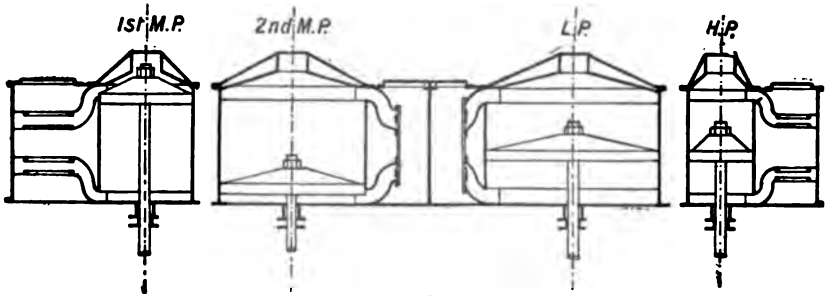
Four-crank engines having two low-pressure cylinders, one high-pressure and one medium-pressure cylinder, each acting on its own crank, are now common enough (Nos. 3 and 4, p. 100); some made so on account of size, and others in order to be of short stroke without a piston of too large diameter for such quick running.

Quadruple-Expansion Engines.—These engines are a further development of the compound principle, whereby steam is expanded through a sequence of four cylinders. The amount of power to be gained by using steam above 150 lbs. pressure is small theoretically, and if, in practice, from the losses which inevitably occur from leakage, &c., there is every reason to anticipate no great practical gain. Again, if to use these higher pressures an additional cylinder is required, some of the gain will be absorbed by increase of friction, &c. No doubt an engine, constructed on plan No. 1, p. 102, using steam of, say, 180 lbs., would compare favourably with one on plan No. 3, p. 99, using steam of, say, 150 lbs., because in each case there are four cylinders on two cranks; but that it would be more economical than one on plan No. 4, p. 99, using steam at 150 lbs. is very problematical. Plan No. 2, on p. 100, may be taken as the same as plan No. 2, on p. 102, so far as size of cylinders is concerned, for there are three cylinders on the main columns of one size and having over each a cylinder of equal size. The steam enters the first of these smaller cylinders and exhausts into the next two; these, in their turn, exhaust into the middle one of the larger cylinders, which, again, exhausts into the other two larger ones, and so forms a quadruple-expansion engine with only two sizes of cylinders.

The same causes that operated to produce the triple-expansion engine have brought out the quadruple. There was, however, the suspicion of an additional cause in this case from the fact that it was first adopted by those engineers who were slow to accept the triple engine as a solution of the existing engineering difficulties; these having been late in the field compensated themselves by adopting the more advanced plan. It cannot, however, be

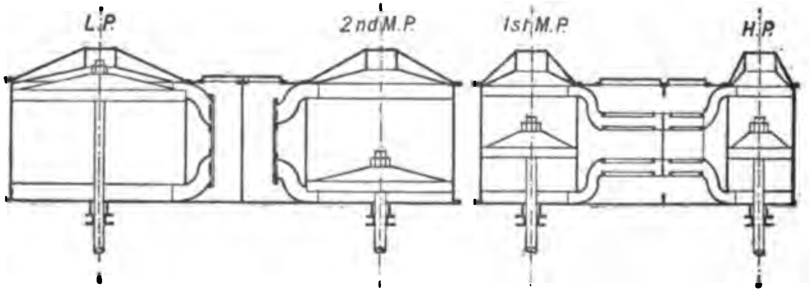


No. 1. *No. 2.*



No. 3.

Four Sets of Valve-Gear.



No. 4.

Two Sets of Valve-Gear.

Fig. 23.—Various Arrangements of the Cylinders of Quadruple-Expansion Two-, Three-, and Four-Crank Engines.

claimed, apart from space occupied, that the two-crank quadruple is so good an engine from the mechanical point of view as the three-crank triple; and although it is claimed that this form of quadruple consumes less coal per I.H.P. per hour than a corresponding triple engine with three cranks, it is very doubtful if its mechanical efficiency is so high, or the consumption per voyage in similar ships is materially less. For the same power the low-pressure cylinder of each engine will be practically the same size, especially if economy of fuel is to enter into consideration. The fourth cylinder with its piston, valves, &c., will cause a distinct addition to the resistance of the engine and a consequent loss of useful work. The objections to increased boiler pressure hold good in this as in other cases, so that losses in practice are likely to arise from it. On the other hand, large increases in boiler pressure are not advisable with a triple engine, and as with the water-tube boiler very great increase in boiler pressure is now possible without any serious risks, the quadruple engine may to a large extent take the place of the triple engine, just as the triple did of the compound engine, especially in the better form of a three-crank or four-crank engine.

The two-crank quadruple was an obvious arrangement of the type at the outset, and it is still obviously commendable for quadrupling the old ordinary compound engines. It had, however, in addition to the objections above mentioned, that appertaining to the four-cylinder ordinary compound—viz., want of accessibility to the pistons of the lower cylinders and extra cost of examining, overhauling, and repairing the engines. The next arrangement to suggest itself would be the addition of a cylinder to a three-crank triple expansion engine, placing it above the original high pressure. This, too, however, is tainted with the same fault. Hence to obtain the most satisfactory quadruple it is better to face at once all the difficulties and employ four cranks, having a cylinder to each; with this arrangement the engine can be well balanced by placing the first and second cranks opposite to one another, the second and third cranks at right angles to one another, and the third and fourth cranks opposite to one another. When this is done one set of link motion is sufficient for the first two cylinders and another set for the second two cylinders.

When, however, the engines are so large as to require two low-pressure cylinders, it is then better to have two high-pressure cylinders, not only as the engine is thereby better balanced but difficulties of construction, owing to the large size of the cylinders, and the high pressure of the steam are avoided. When this plan is adopted the most symmetrical design is obtained by putting the two low-pressures side by side with the two high-pressures above them, first medium pressure on the fore side of the low-pressure cylinders, and the second medium pressure on the after side. It is then easy to divide the work evenly between the four cranks. Mr. Mudd, however, preferred to have five cranks.

The consumption of fuel with a good quadruple-expansion engine with a boiler pressure of 200 lbs. per square inch should not exceed 1¼ lbs. of Welsh coal per I.H.P. per hour. Even better results than this are stated to have been obtained in practice, but seeing that the water consumption as measured by the indicator cards of these engines has been nearly 12 lbs. per I.H.P. per hour, the coal consumption is likely to have been higher than above named. Table viii. gives the water consumption of various triple and quadruple engines, by which it will be seen that the saving of the latter over the former is about 7·7 per cent. It must not, however, be supposed that the percentage of I.H.P. usefully employed is the same in all these engines.

TABLE VIII.—RESULTS OF STEAM TRIALS WITH VARIOUS MARINE ENGINES.

NAME OF VESSEL.	Description of Engines.	Diam. of Cylinders.	Stroke.	Pres. at Engine Room.	Rate of Expansion.	Revolutions per Minute.	Indicated H.P.	Speed of Vessel.	Ind. Consumpt. Lbs. of Water per Stroke.	Ind. Consumpt. Lbs. per I.H.P. per Hour.
		Inches.	Inches.	Lbs.				Knots.		
S.S. Northern, - -	Ord. comp. engs.,	26 56	60	115	7·73	57·5	1242	9·25	2·57	15·4
,, Kovno, - - -	Do.,	25 50	45	90	6·6	73	809	10·6	1·473	16·6
,, Bolama, - - -	Do.,	24 45	33	80	6·2	82	562	11·23	·867	16·8
,, Draco, - - -	2-Crank trip. engs.,	21 32 56	36	110	11·97	73	618	10·5	·975	13·8
,, Finland, - - -	Do.,	20 32 56	36	150	12·2	72	806	9·0	1·136	12·88
,, Rosario, - - -	3-Crank trip. engs.,	20½ 33 58	36	150	15·1	62	664	9·0	1·007	11·3
,, Don, - - -	Do.,	40 63 96	54	145	8·73	72	4702	16·82	7·44	13·67
,, Ariosto, - - -	Do.,	30 46 75	45	148	9·92	79	2601	14·6	4·05	14·7
,, Victoria, - -	Do.,	23½ 35 57	33	148	7·94	105	1522	14·9	1·747	14·4
,, Queen, - - -	Do.,	12¾ 20 32	22	150	8·88	132	324	11·07	·261	12·75
,, Pegu, - - -	Quad. comp. engs.,	25¾ 38½ 51½ 75	51	179	14·16	59·8	1856	...	2·775	10·73
,, Righi, - - -	Do.,	17 24 35 50	27	184	15·17	72	683	10·5	·881	12·0
,, European, - -	Do.,	14 21 29 44	27	175	16·03	82	442	11·2	·537	11·9
,, Polo, - - -	Do.,	12¾ 18¼ 27 39	24	200	15·01	98	495	11·3	·467	11·1
H.M.S. Magpie, - -	3-Crank trip. engs.,	20 30 45	24	138	6·93	189·5	1358	14·25	·947	15·9
,, Philomel P., -	* Do.,	30¼ 45 68	33	153	5·84	167·2	4085	18·43	3·29	16·2
,, Andromache S.,	* Do.,	33½ 49 74	39	152	5·74	136·9	4771	19·5	4·51	15·5
,, Endymion P., -	* Do.,	40 59 88	51	150	5·7	100	5338	20·96	6·28	14·1

* Twin-screws. One engine only.

Cylinders as found in Actual Practice.—(*a*) For compound engines whose power does not demand a low-pressure cylinder of such dimensions as are inconvenient, or beyond the capabilities of manufacture, and when the rate of expansion is not such as to demand special arrangements, two cylinders may be used. In actual practice it is not usual to make cylinders over 100 inches in diameter, and many engineers prefer to limit the size to 90 inches, although engines have been made with cylinders over 120 inches. The rate of expansion may be taken at one-tenth of the boiler pressure (or about one-twelfth the *absolute* pressure), to work economically at full speed. Therefore, when the diameter of the low-pressure cylinder does not exceed 100 inches, and the boiler pressure 70 lbs., the ratio of the low-pressure to the high-pressure cylinder should be 3·5; for a boiler pressure of 80 lbs., 3·75; for 90 lbs., 4·0; for 100 lbs., 4·5. If these proportions are adhered to, there will be no need of an expansion-valve to either cylinder. If, however, to avoid "drop" the ratio be reduced, an expansion-valve should be fitted to the high-pressure cylinder.

Where economy of fuel is not of first importance, but rather a large power, the ratio of cylinder capacities may, with advantage, be decreased, so that with a boiler pressure of 100 lbs. it may be 3·75 to 4.

(*b*) The two-cylinder compound engine having direct expansion in its most perfect form, is arranged with the cylinders side-by-side, and each piston operating on an independent crank opposite the other. The ratio of the cylinders has been generally smaller in this case than that given above. When the cylinders are placed one above the other, as in the single-crank engine, there is no *necessity* to divide the work equally, although it is better to do so generally; the chief difficulty in the way of effecting this by means of the cut-off, arises from the fact that both valves are usually worked by the same eccentrics and gear, and the cut-off must be practically the same in both. Hence, if any variation in the division of the work is required, it must be done by means of a separate expansion-valve on the high-pressure cylinder, or by the ratio of the cylinders. From practical considerations the ratio is generally 4; but when the steam pressure exceeds 90 lbs. absolute, it is better to be 4·5; and for 100 lbs., 5·0.

(*c*) When the power requires that the low-pressure cylinder shall be more than 100 inches diameter, the three-cylinder engine with two low-pressure cylinders, or the double "tandem" engine, having two low-pressure and two high-pressure cylinders, should be adopted. In the latter case, the same proportions as given for a single-crank engine will be observed. In the former case, the ratio of the combined capacity of the two low-pressure cylinders to that of the high-pressure may be 3·0 for steam pressure of 85 lbs. absolute, for 95 lbs. absolute 3·6, and for 105 lbs. absolute 3·8, and for 115 lbs. absolute 4 for naval engines and a ratio of 8 per cent. larger for mercantile ones.

(*d*) When the pressure of steam employed exceeds 115 lbs. absolute it is advisable to employ three cylinders, through each of which the steam expands in turn. The ratio of the low-pressure to high-pressure cylinder in this system *should be* 5, when the steam pressure is 155 lbs. absolute; when 165 lbs. absolute, 5·4; when 175 lbs. absolute, 5·8; when 185 lbs. absolute, 6·2; when 195 lbs. absolute, 6·6; when 205 lbs. absolute, 7·0; when 215 lbs. absolute, 7·4; and when 225 lbs. absolute, 7·8. In naval and short voyage express steamers, the ratios may be decreased so that 7·8 will be right for 240 absolute, and 5 for 170 lbs. absolute, &c. The introduction of the triple-compound engine has admitted of ships being propelled at higher rates of speed than formerly obtained without exceeding the consumption of fuel of similar ships fitted with ordinary compound engines; in such cases the higher power to obtain the speed has been developed by decreasing the rate of expansion, the low-pressure cylinder being only 6·2 times the capacity of the high-pressure, with a working pressure of 200 lbs. absolute. It is now a very general and convenient practice to proportion the cylinders so as to make the diameter of the low-pressure cylinder equal to the sum of the diameters of the high-pressure and medium-pressure cylinders; hence—

Diameter of M.P. cylinder = 1·5 diameter of H.P. cylinder.
Diameter of L.P. cylinder = 2·5 diameter of H.P. cylinder.

In this case the ratio of L.P. to H.P. is 6·25; the ratio of M.P. to H.P. is 2·25; and ratio of L.P. to M.P. is 2·78.

(*e*) When the pressure of steam employed exceeds 190 lbs. absolute, four cylinders should be employed with the steam expanding through each successively; this system is the quadruple-compound, and the ratio of L.P. to H.P. should be at least 7·5, and if economy of fuel is of prime consideration, it should be 8; then the ratio of first M.P. to H.P. should be 1·8, that of second M.P. to first M.P. 2, and that of L.P. to second M.P. 2·2.

Turbo-Motors.—The success obtained by the Hon. Charles Parsons with his patent turbo-motors on board ship, notably in the cases of the screw steamers "King Edward" and "Queen Alexandra" and H.M.S. "Viper," turned the attention of marine engineers to this form; since then a considerable number of cross-channel steamers have been fitted with them. The trials of H.M.S. "Amethyst," 14,000 I.H.P., were considered so satisfactory that the Admiralty have adopted this propeller for all classes of ships. Notwithstanding that all that is said of them by their supporters may not be accepted without question by engineers generally, it must be admitted by all that these motors possess some distinct advantages over the ordinary engine, especially in economy of space vertically and minimising of the attention of the attendants so as to render them very acceptable for particular service, and, no doubt, with the improvements that are bound to follow their use on ship board, they must become there, in course of time, formidable competitors of the ordinary steam engine. At present the very high speed of revolution necessary to obtain the maximum efficiency, and the inability to reverse by themselves without extraneous help, seriously handicaps their general use for marine propulsion.

Great credit and praise is due to Mr. Parsons for the persistent way in which he has developed his original ideas, overcome difficulty after difficulty, generally by most ingenious methods, and brought his machine to such a state of perfection as to permit of its employment in a service where continuity of use and freedom from risk of breakdown are absolutely necessary.

The first steam engine ever made (Hero's) was a rotatory one, and although differing from Parsons, it was designed on a most important principle common to both—viz., the obtaining of a rotatory motion direct without the interposition of cylinders, cranks, rods, &c. For more than a century attempts have been made by almost every eminent engineer to perfect a rotatory engine of some kind, most of which have shown considerable ingenuity, as also a singular blindness to some of the first principles of practical engineering. They have also nearly always involved moving parts which, sooner or later, must have become worn and out of order with a consequent decrease in efficiency proceeding at a geometrical rate of progression. Other attempts on lines somewhat similar to those of Mr. Parsons have been made from time to time, but at present the chief competitors in the field are the De Laval (fig. 24), manufactured in this country by Messrs. Greenwood & Batley; the Rateau, invented by the French professor of that name; the Curtis, of American origin; and the Schultz, made in Germany.*

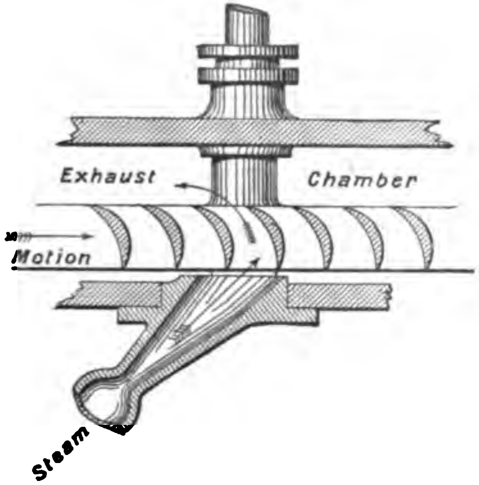


Fig. 24.—De Laval Turbo-Motor.

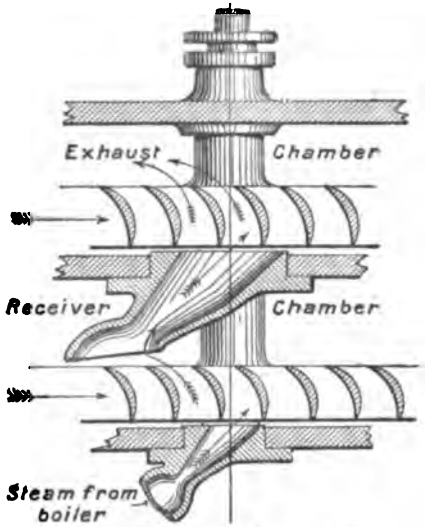


Fig. 24a.—Compound Turbo-Motor (De Laval system).

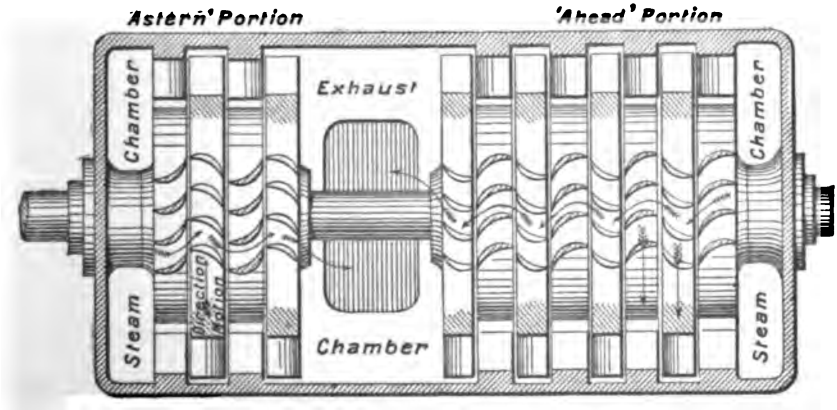


Fig. 24b.—Parsons' Turbo-Motor (Parallel Flow). Elemental Form.

Fig. 24a shows a system of compounding of the De Laval motor, whereby the expansion of the steam, instead of being by means of one set of nozzles, as shown in fig. 24, is partly through one set and continued in a second set as shown. In this way the rate of revolution is reduced, and possibly a higher rate of efficiency obtained. The Curtis turbine is on this principle.

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* *Vide The Theory of the Steam Turbine*, by A. Jude. Griffin & Co., Ltd.

Fig. 24*b* shows the arrangement of the vanes in a parallel flow in Parsons' turbo-motor. It consists of a series of discs keyed to one shaft, and having fitted in their edges vanes formed as shown. These discs revolve in a chamber having collars projecting inwards and carrying similar vanes to those on the discs, but set roughly at right angles to those on the discs. Each disc has, therefore, in front of it a set of vanes fixed and arranged to guide the steam so that its flow is normal to the disc vanes. In Mr. Parsons' original motors these vanes were flat—that is, straight in section—but now they are curved as shown in fig. 24*b*, so that the steam enters axially, and is gently diverted so as to pass normally to the entering edge of the vanes on the first disc, which are also curved so as to receive the steam squarely and deliver it at exhaust in an easy trail, to be caught by the next set of fixed vanes and diverted to those of the second disc, and so on. Mr. Parsons has compounded his motors by fitting a second chamber with discs, &c., of larger diameter than those in what may be termed the high-pressure motor, and thereby obtained a further expansion from the steam before entering the condenser.

There is, of course, no limit to this system, as by extending in this way a quadruple, quintuple, or sextuple motor can easily be designed.*

The ingenious way in which Mr. Parsons gets over the end-thrust, vibration, general balancing, &c., need not be gone into here, but it may be stated that reversing in the case of the above ships is obtained by fitting a shorter turbo-motor in the rear of the low-pressure (when compounded) headgoing motors, the vanes of this motor being, of course, reversed in direction; the propellers are, of course, reversed by steam being shut off from the headgoing portion and turned on to the sterngoing. In the ordinary ships there are three propellers on three separate shafts, all of which are worked by a compound set of turbo-motors; and in the case of some of H.M. ships the middle propeller is worked by an ordinary triple-compound reciprocating engine, which may not only be used for the sterngoing work, but likewise for cruising speeds, as at present the Parsons motor loses its efficiency very considerably with the decrease in power developed, so that when running slow it is very uneconomical. In ships of large size and high power, such as the largest of the Cunard steamers, there are four propellers on four shafts.

CHAPTER VI.

EXPANSION OF STEAM, MEAN PRESSURE, ETC.

THE diameter of the steam cylinder depends on the piston speed and mean pressure permissible in obtaining the required I.H.P.

The Mean Pressure depends on the initial pressure of the steam and the rate of expansion observed in working, and may be calculated from data given in Table ix.

In Table ix. p_1 is the *absolute initial pressure*; p_m the absolute mean pressure; r the rate of expansion; $\frac{1}{r}$ the cut-off. The calculations are based on the supposition that the steam is *moderately moist*, and expands in accordance with Boyle & Marriott's law ($p\,v$ = constant), so that the pressure varies inversely with the volume.

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* *Vide* Appendix N.

Then,

$$\text{The mean pressure} = p_1 \frac{1 + \text{hyp. log } r.}{r}$$

The hyperbolic logarithm of a number may be found by multiplying the common logarithm of that number by 2·302585.

$$\text{The terminal pressure} = \frac{p_1}{r}$$

Therefore,

(1) Ratio of mean pressure to terminal pressure $= \frac{rp_m}{p_1}$.

(2) Ratio of terminal pressure to mean pressure $= \frac{p_1}{rp_m}$.

(3) Ratio of maximum pressure to mean pressure $= \frac{p_1}{p_m}$.

TABLE IX.—Steam used Expansively.

r	$\frac{1}{r}$	$\frac{rp_m}{p_1}$	$\frac{p_1}{rp_m}$	$\frac{p_1}{p_m}$	$\frac{p_m}{p_1}$	$\frac{p_m}{p_1}$ Dry.
20	0·050	4·00	0·250	5·00	0·1998	0·186
18	0·055	3·89	0·256	4·63	0·2161	...
16	0·062	3·77	0·265	4·24	0·2358	...
15	0·066	3·71	0·269	4·05	0·2472	...
14	0·071	3·64	0·275	3·85	0·2599	...
13·33	0·075	3·59	0·279	3·72	0·2690	0·254
13	0·077	3·56	0·280	3·65	0·2742	...
12	0·083	3·48	0·287	3·44	0·2904	...
11	0·091	3·40	0·294	3·24	0·3089	...
10	0·100	3·30	0·303	3·03	0·3303	0·314
9	0·111	3·20	0·312	2·81	0·3552	...
8	0·125	3·08	0·321	2·60	0·3849	0·370
7	0·143	2·95	0·339	2·37	0·4210	...
6·66	0·150	2·90	0·345	2·30	0·4347	0·417
6·00	0·166	2·79	0·360	2·15	0·4653	...
5·71	0·175	2·74	0·364	2·08	0·4807	...
5·00	0·200	2·61	0·383	1·92	0·5218	0·506
4·44	0·225	2·50	0·400	1·78	0·5608	...
4·00	0·250	2·39	0·419	1·68	0·5965	0·582
3·63	0·275	2 29	0·437	1·58	0·6308	...
3·33	0·300	2·20	0·454	1·51	0·6615	0·648
3·00	0·333	2·10	0·476	1·43	0·6993	...
2·86	0·350	2·05	0·488	1·39	0·7171	0·707
2·66	0·375	1·98	0·505	1·34	0·7440	...
2·50	0·400	1·91	0·523	1·31	0·7664	0·756
2·22	0·450	1·80	0·556	1·24	0·8095	0·800
2·00	0·500	1·69	0·591	1·18	0·8465	0·840
1·82	0·550	1·60	0·626	1·14	0·8786	0·874
1·66	0·600	1·51	0·662	1·10	0·9066	0·900
1·60	0·625	1·47	0·680	1·09	0·9187	...
1·54	0·650	1·43	0·699	1·07	0·9292	0·926
1·48	0·675	1·39	0·718	1·06	0·9405	...

When steam expands in accordance with the law $p v =$ constant, the curve drawn through the extremities of ordinates representing the pressure at any position of the piston is a hyperbola. The mean height of such a system of ordinates is found by the formula given above; this mean height will represent the mean pressure.

The mean pressure may be obtained without the aid of logarithms, by resorting to arithmetical calculation of the ordinates, and finding the mean by the method explained in Chapter ii.

Example.—Initial pressure 80 lbs., rate of expansion 5. Suppose the length of stroke divided into ten equal parts by points 1, 2, 3, . . . 9, 10. The cut-off is $\frac{1}{5}$, or two-tenths the stroke.

The pressure at commence of stroke is				80 lbs.
„	1 tenth	„		80 „
„	2 „	„		80 „
„	3 „	„	$\frac{2}{3}$ of 80 or	53·33 „
„	4 „	„	$\frac{2}{4}$ „	40·00 „
„	5 „	„	$\frac{2}{5}$ „	32·00 „
„	6 „	„	$\frac{2}{6}$ „	26·66 „
„	7 „	„	$\frac{2}{7}$ „	22·86 „
„	8 „	„	$\frac{2}{8}$ „	20·00 „
„	9 „	„	$\frac{2}{9}$ „	17·78 „
„	10 „	„	$\frac{2}{10}$ „	16·00 „

$$p_m = \frac{1}{10}\left(\frac{80 + 16}{2} + 80 + 80 + 53{\cdot}33 + 40 + 32 + 26{\cdot}66 + 22{\cdot}86 + 20 + 17{\cdot}78\right)$$
$$= 42{\cdot}063 \text{ lbs.}$$

By reference to Table IX., for the rate of expansion 5, $\frac{p_m}{p^1} = 0{\cdot}5218$, and for 80 lbs. pressure $p_m = 41{\cdot}744$ lbs., or about $\frac{3}{4}$ per cent. less than that given by the summation above, the excess of which is due to the small number of points of observed pressure.

Graphic Method.—Professor Rankine has given* a geometric method of ascertaining the mean pressure, which is of interest and value, and which first appeared in the *Engineer*. Draw a straight line, A B, of definite length, produce it to A C, so that $A C = \frac{A B}{4}$. Through A draw A D at right angles to C A B. With C as centre, and C B as radius, draw the arc of a circle, cutting A D at D. Then if $\frac{D A}{D E}$ is the rate of expansion $\frac{p_m}{p_1} = \frac{E F}{A B}$.

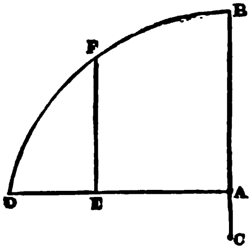

Fig. 25.

To suit this diagram for actual use, A B should be taken of such a length as is convenient for scale measurement, say 4 inches; A D should be divided into ten parts and subdivided into quarters; through the divisions faint lines should be drawn parallel to A B. Scales should be constructed 4 inches long, suitable for the usual pressures coming under consideration.

The object of columns 3, 4, in Table ix., is that mean pressure and initial pressure may be easily determined from terminal pressures, when the rate of expansion is known; or when initial and mean pressures are known, the rate of expansion may be found. Column 5 is given to show the relation between maximum and mean pressures at the various rates of expansion.

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* Rankine, *Rules and Tables*, p. 291

Dry Steam.—When steam is dried by superheating, so that it is surcharged with heat, and is capable of very considerable expansion without liquefaction taking place, it expands according to the law of perfect gases, and then

$$\frac{p_m}{p_1} = \frac{17 - 16r^{-\frac{1}{16}}}{r}$$

$r^{-\frac{1}{16}}$ may be found by extracting the square root of $\frac{1}{r}$ four times.

$$\left(r^{-\frac{1}{16}} = \sqrt{\sqrt{\sqrt{\sqrt{\frac{1}{r}}}}}\right)$$

Column 6 gives the value of $\frac{p_m}{p_1}$ as calculated from the above formula by the late Professor Rankine. It will be seen that, except at very high rates of expansion, there is no very great difference between the ratio as calculated by this method and by the method for moderately moist steam.

Clearance.—In practice the mean pressure in the cylinder is very materially affected by what is called *clearance*. However accurately the engine is constructed, there is always a commencement of the stroke a space between the piston and cut-off valve, made up of the part of the cylinder between the piston and the cover or cylinder end, and the passage between valve face and cylinder; this is called the *clearance*. Supposing this space is equal to one-tenth of the capacity of the cylinder, and the cut-off is at two-tenths the stroke, the *effective* cut-off is not two-tenths, but something more, due to the fact that the expansion of a volume of steam equalling three-tenths the capacity of cylinder is being effected, instead of that of a volume of two-tenths. This practically amounts to making the cylinder 10 per cent. longer, and cutting off at three-tenths the stroke without clearance. It is, therefore, customary to speak of the clearance as amounting to such a proportion of the stroke.

To allow for the effect which the *clearance* will have when steam expands in a cylinder, let r be the nominal rate of expansion as before, and r_1 be the *actual* rate allowing for clearance, c the clearance as a fraction of the cylinder capacity. Then

$$\frac{1}{r_1} = \frac{\frac{1}{r} + c}{1 + c} \quad \text{and } r_1 = r\,\frac{1 + c}{1 + cr} \quad - \quad - \quad - \qquad (A)$$

$\frac{1}{r} + c$ being the volume of steam at cut-off between the piston and the cut-off valve, and which expands to the volume $1 + c$ at the end of the stroke. If there is no cushioning of the steam before admission, then the whole of the space $\frac{1}{r} + c$ must be filled at each stroke with fresh steam.

Then the *real mean absolute pressure* will be

$$p'_m - c(p_1 - p'_m) \quad - \quad - \quad - \quad - \qquad (B)$$

p' is the mean pressure obtained by means of Table IX., the *actual* rate of expansion being taken, and p_1 is the absolute initial pressure. If, however, there is sufficient cushioning to fill the clearance space with steam at the initial pressure, the volume of steam used at each stroke will be only that swept by the piston at cut-off and equal to $\frac{1}{r}$.

Compression or Cushioning.—There will be an increase of back pressure caused by this cushioning, and its effect on the mean pressure is as follows:—

Let p'_m be the mean absolute pressure due to the *effective* cut-off $\frac{1}{r_1}$; p_0 the absolute back pressure; c the clearance; and p_1 the absolute initial pressure as before.

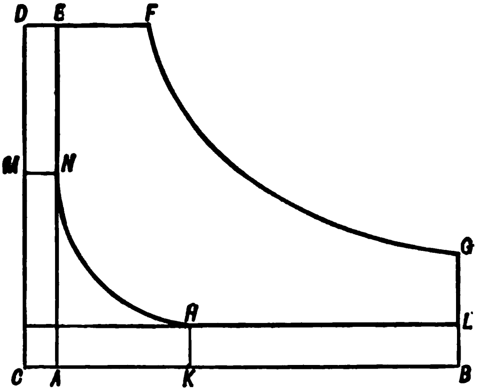

Fig. 26.

Fig. 26 is the indicator-diagram of an engine working under these conditions. A B is the stroke, A C the clearance, E F the nominal cut-off, and D F the effective. The actual rate of expansion is therefore $\frac{\text{C B}}{\text{D F}}$. C D represents the initial pressure and H K the back pressure. Cushioning commences at K with pressure p_0, and at A the pressure is p_1.

The figure C D E H K is the diagram due to the expansion of steam of a volume equalling c at a pressure p_1 to a pressure p_0, so that the rate of its expansion is $\frac{\text{C K}}{\text{D E}}$. Now $p_1 \times \text{D E} = p_0 \times \text{C K}$, and, therefore,

$$\frac{p_1}{p_0} = \frac{\text{C K}}{\text{D E}}.$$

Since p_1 and p_0 are both known the rate of expansion is known, and by referring to Table IX. the mean pressure p''_m due to this rate of expansion is found.

The area H E F G $=$ area C D F G B $-$ (area C D E H K $+$ area H K B)
" C D F G B $= p'_m \times (\text{A B} + \text{A C}) = p'_m(1 + c)$.
" H K B $= p_0(\text{A B} - \text{A K}) = p_0\left(1 - \left(c\frac{p_1}{p_0} - c\right)\right)$.
" C D E H K $= p''_m(\text{C K}) = p''_m\left(\frac{p_1}{p_0}c\right)$.
" H E F G $= p_m \times 1$, or the effective mean pressure.

Therefore, the effective mean pressure

$$= p'_m (1 + c) - \left\{1 - c\left(\frac{p_1}{p_0} - 1\right)\right\} p_0 - p''_m \left(\frac{p_1}{p_0} - c\right)$$

$$= p'_m \times c\,(p'_m - p_0) + cp_1 \left(1 - \frac{p''_m}{p_0}\right) - p_0$$

$$= (p'_m - p_0)(1 + c) + cp_1 \left(1 - \frac{p''_m}{p_0}\right). \quad - \quad - \quad - \quad (C)$$

General Effect of Clearance and Cushioning.—Let p', the absolute initial pressure, be represented in Fig. 26 by C D, p_0 the back pressure by B L, A B the length of stroke, A C the clearance c, A K the compression x, E F the nominal cut-off, r the nominal rate of expansion, r_1 the real rate of expansion, &c., &c., as before. p'_m the mean pressure due to an initial pressure p', and a rate of expansion r_1; p_m the real mean pressure of N E F G L H. As before

$$r_1 = \frac{1 + c}{\dfrac{1}{r} + c} = r\frac{1 + c}{1 + cr}. \quad - \quad - \quad - \quad (1)$$

Since the steam at point K is shut up in a space $x + c$, and is compressed into a space c, the *rate of compression* is $\dfrac{x + c}{c}$; and the pressure after compression at N is $p^c = \dfrac{x + c}{c} p_0$, and represented by A N or C M; let p^c_m be the mean pressure of the figure M N H K C, which is that due to a pressure $\dfrac{x + c}{c} p_0$, and a rate of expansion $\dfrac{x + c}{c}$.

The area NEFGLH = CDFGB - DENM - MNHKC - KHLB.

„ NEFGLH $= p_m \times 1.$

„ CDFGB $= p'_m (1 + c).$

„ DENM $= \left(p' - p_0 \dfrac{x + c}{c}\right) c = (p' - p_0)\,c - p_0 x.$

„ MNHKC $= p^c_m (x + c).$

„ KHLB $= p_0 (1 - x).$

Therefore,

$$\begin{aligned} p_m &= p'_m (1 + c) - \{(p' - p_0)\,c - p_0 x\} - p^c_m (x + c) - p_0 (1 - x) \\ &= p'_m (1 + c) - p'c - p_0 (1 - 2x - c) - p^c_m (x + c). \quad - \quad (2) \end{aligned}$$

Example I.—To find the effective mean pressure in a cylinder having a clearance space equal to one-seventh its capacity, the initial pressure 80 lbs. absolute, the back pressure 10 lbs. absolute, and the nominal cut-off $\frac{1}{5}$ the stroke:

(1) If no compression

$$r = 5\,\frac{1 + \frac{1}{7}}{1 + \frac{5}{7}} = 5 \times \tfrac{8}{12} = 3{\cdot}33.$$

By reference to Table ix. $\dfrac{p_m}{p_1} = 0{\cdot}6615$ for a rate of expansion $= 3{\cdot}33$.

Then

$$p'_m = 80 \times 0{\cdot}6615 = 52{\cdot}92 \text{ lbs.},$$

and

The effective mean pressure $= 52{\cdot}92 - 10 = 42{\cdot}92$ lbs.

(2) If full compression to 80 lbs.: Here rate of compression

$$r_0 = \frac{p_1}{p_0} = \frac{80}{10} = 8.$$

Therefore,

$$\frac{p^{\circ}_m}{p_0} = \frac{r_0 p^{\circ}_m}{p_1} = 3{\cdot}08 \text{ (Table ix.)}$$

$$p'_m = 52{\cdot}92 \text{ lbs. as before.}$$

Then the effective mean pressure by formula (C), p. 113,

$$= (52{\cdot}92 - 10)\ (1 + \tfrac{1}{7}) + \tfrac{80}{7}\ (1 - 3{\cdot}08).$$

$$= \tfrac{8}{7} \times 42{\cdot}92 - \frac{166{\cdot}4}{7} = 25{\cdot}28 \text{ lbs.}$$

If there was no clearance, the effective mean pressure would be 41·74 – 10, or 31·74 lbs.

The steam used in the case (2) is the same as if there had been no clearance, and as the effective mean pressure was only 25·28 lbs., there is a *loss due to clearance* of 6·46 lbs., or 20 per cent. In case (1) the quantity of steam used is $\frac{12}{35}$ the volume of the cylinder per stroke, or one-seventh of the volume in excess of the quantity with no clearance, so that with this increase of steam, if there was no clearance and the rate of expansion 5, there should be an increase in the work done, and that increased work will be to the work done by the smaller quantity of steam as 12 is to 7.

The equivalent mean effective pressure is then $\frac{12}{7}$ of 31·74, or 54·41, as against 42·92 lbs. which was obtained, showing a loss of 11·49 lbs., or 21 per cent.

The example given is a very extreme case, and such as would be rarely found in practice. The effect of clearance in the high-pressure cylinder of a compound engine may be seen in the following:—

Example II.—The nominal rate of expansion is 2, the initial absolute pressure 90 lbs., and the absolute back pressure $22\frac{1}{2}$ lbs., the clearance being one-ninth the capacity of the cylinder.

(1) No compression:

$$r = 2\frac{1 + \frac{1}{9}}{1 + \frac{2}{9}} = 1{\cdot}82.$$

By reference to Table IX., $\dfrac{p_m}{p_1} = {\cdot}8786$ for a rate of expansion of 1·82

Then $$p_m = 90 \times 0{\cdot}876 = 79 \text{ lbs.}$$

Mean effective pressure $= 79 - 22{\cdot}5$, or 56·5 lbs.

The equivalent mean pressure when $\frac{1}{2} + \frac{1}{9}$ or $\frac{11}{18}$ of the volume of the cylinder of steam is used will be $\frac{11}{9}$ of 53·68 lbs., or 65·61 lbs., showing a loss by clearance of 13·88 per cent.

(2) If full compression to 90 lbs.:

Here $\dfrac{p_1}{p_0} = 4$, which is the *rate of compression;* so that

$$\frac{p^{\circ}_m}{p} = \frac{r_0 p^{\circ}_m}{p_1} = 2{\cdot}39 \quad \text{- - - Table ix.}$$

The effective mean pressure by formula (C), p. 113,

$$= (79 - 22·5)(1 + \tfrac{1}{9}) + \tfrac{90}{9}(1 - 2·39)$$
$$= 62·77 - 13·9 = 48·87 \text{ lbs.}$$

Thus showing a loss of 5·81 lbs., or 10·8 per cent. only. The loss from the clearance in a compound is not so serious as in the expansive engine, and in a triple or quadruple engine is of no consequence, as the steam in the former (which has passed from the high-pressure cylinder without giving out its full work), will do more work in the medium-pressure and low-pressure cylinders; whereas, with the expansive engine, the exhaust steam passes direct to the condenser at a higher pressure than when there is no clearance. Further, since the cut-off in an expansive engine is much earlier than in a compound, and the clearance from practical considerations is very much the same, the ratio of clearance to volume at cut-off will be much higher in the former than in the latter. Considerable loss is, however, experienced in compound, triple, and quadruple engines if the clearance in the low-pressure cylinder is large.

The beneficial effect of cushioning is seen in both the preceding examples, but its value is greater still when the cut-off in the high-pressure cylinder is somewhat earlier, as may be seen by the following:—

Example III.—The cut-off in the cylinder of example (2) is altered to ¼ the stroke, so that the nominal rate of expansion is 4.

(1) No compression:

$$r = 4\frac{1 + \frac{1}{9}}{1 + \frac{4}{9}} = 3.$$

Then $$p_m = 62·1 \text{ lbs.}$$

and the effective mean pressure = 62·1 − 22·5 = 29·6 lbs.

The equivalent mean pressure due to the amount of steam used is now $\frac{13}{9}$ of 31·185, or 45 lbs., thus showing a loss of 12 per cent.

(2) If full compression to 90 lbs.:

The effective mean pressure by formula C

$$= (62·1 - 22·5)\tfrac{10}{9} + \tfrac{90}{9}(1 - 2·39)$$
$$= 44 - 13·9 = 30·1 \text{ lbs.}$$

Thus showing a loss of 1·085 lbs., or 3·4 per cent. only.

The economy effected by working with a considerable amount of cushioning is, therefore, very appreciable, and practice has proved the correctness of this.

In actual practice, however, it seldom happens that so much cushioning can be effected as to fill the clearance space with steam of pressure equal to that of the entering steam; but still even what is conveniently obtained materially adds to the economic working of the engine. It must not, however, be forgotten that the effective mean pressure is considerably reduced by cushioning.

Mean Pressure in a Compound Engine.—If the effective mean pressure in the high-pressure cylinder of a compound engine be divided by the ratio of capacity of low-pressure to that of the high-pressure cylinder, the quotient represents the mean pressure necessary to do the same work in the low-pressure cylinder as is effected in the high-pressure cylinder. If this be added to the effective mean pressure in the low-pressure cylinder, the sum will be the mean pressure necessary to obtain from the low-pressure cylinder alone, the whole work done by both cylinders, and may be called the *equiva-*

lent mean pressure. If there be no loss of mean pressure, owing to drop in the receiver, or other cause, this equivalent mean pressure will be the same as the effective mean pressure obtained by the steam expanding in one cylinder at the same rate as the total expansion effected in both cylinders of the compound engine. In the two-cylinder receiver form of compound engine, there is sometimes a considerable fall in pressure from the release point to the exhaust, owing to the low pressure maintained in the receiver.

(1) Two-cylinder receiver compound engine.

Let p_1 be the initial pressure, p_0 the back pressure in the low-pressure cylinder, p_r the pressure in the receiver and back pressure in the high-pressure cylinder; R the ratio of cylinder capacities, r the total rate of expansion, r_1 the rate of expansion in the high-pressure cylinder, and r_n that in the low-pressure cylinder; p'_m the mean pressure due to an initial pressure, p^1, and a rate of expansion, r_1 : p''_m, the mean pressure due to an initial pressure p_r, and a rate of expansion r_2, P_m is the mean pressure due to a rate of expansion r, and an initial pressure p_1:

The effective mean pressure in the high-pressure cylinder is then $(p'_m - p_r)$; and that in the low-pressure cylinder is $(p''_m - p_0)$.

Also

$$P_m = p_1 \frac{1 + \text{hyp. log. } r}{r},$$

$$p'_m = p_1 \frac{1 + \text{hyp. log. } r_1}{r_1},$$

$$p''_m = p_r \frac{1 + \text{hyp. log. } r_2}{r_2}.$$

Since the work performed in the engine is supposed to be equally divided between the two cylinders,

$$p'_m - p_r = \text{R}\,(p''_m - p_0). \quad - \quad - \quad - \quad (1)$$

But if there be no loss due to "drop," and the mean pressure in the high-pressure be referred to the low-pressure cylinder, then

$$\frac{p'_m - p_r}{\text{R}} + (p''_m - p_0) = P_m - p_0.$$

By substituting the value of $(p'_m - p_r)$ of (1) in the above

$$\left.\begin{aligned} p''_m - p_0 &= (P_m - p_0)\tfrac{1}{2} \\ \text{and} \quad p'_m - p_r &= (P_m - p_0)\frac{\text{R}}{2} \end{aligned}\right\} \quad - \quad - \quad - \quad (2)$$

Let x be the efficiency of the system, so that $(1 - x)$ is the proportion of loss due to drop. Then

$$\left.\begin{aligned} p''_m - p_0 &= x\,(P_m - p_0)\tfrac{1}{2} \\ \text{and} \quad p'_m - p_r &= x\,(P_m - p_0)\frac{\text{R}}{2} \end{aligned}\right\} \quad - \quad - \quad - \quad (3)$$

To find the *actual* mean pressures when there is loss due to "drop," the value of x must be determined; this may be done by substituting the value of p'_m and p''_m, found by the preceding formulæ; but an approximate value may be found by determining the value of p_r in equation (3); from the value thus found calculate p''_m, and refer the mean pressures of both cylinders to the low-pressure cylinder. If $(P'_m - p_0)$ be the equivalent mean pressure thus found, then, approximately,

$$x = \frac{P'_m - p_0}{P_m - p_0}. \quad - \quad - \quad - \quad - \qquad (4)$$

Example.—To find the mean pressure in a compound engine using steam of 90 lbs. absolute pressure, the total rate of expansion being 7, the ratio of the cylinder capacities 3·5, and the back pressure 4 lbs.

$$P_m = 90 \times 0{\cdot}421 = 37{\cdot}89 \text{ lbs.} \quad - \quad - \quad \text{Table ix.}$$

$$r_1 = 7 \div 3{\cdot}5 = 2.$$

Then

$$p'_m = 90 \times 0{\cdot}8465 = 76{\cdot}18 \text{ lbs.} \quad - \quad - \quad \text{Table ix.}$$

$$p_r = 76{\cdot}18 - \frac{3{\cdot}5}{2}(37{\cdot}89 - 4) = 16{\cdot}88 \text{ lbs.}$$

$$r_2 = \frac{p_r r}{p_1} = \frac{16{\cdot}88 \times 7}{90} = 1{\cdot}313,$$

$$p''_m = 16{\cdot}88 \, \frac{1 + \text{hyp. log. } 1{\cdot}313}{1{\cdot}313} = 16{\cdot}36.$$

That is, the effective mean pressure in high-pressure cylinder is 76·18 − 16·88, or 59·3 lbs., and that in low-pressure cylinder is 16·36 − 4, or 12·36. Referred to low-pressure cylinder alone,

$$P'_m - p_0 = 12{\cdot}36 + \frac{59{\cdot}3}{3{\cdot}5} = 29{\cdot}3 \text{ lbs.}$$

$$P_m - p_0 = 37{\cdot}89 - 4 = 33{\cdot}89 \text{ lbs.}$$

Therefore,

$$x = \frac{29{\cdot}3}{33{\cdot}89} = 0{\cdot}865.$$

Then, actual effective mean pressure in high-pressure cylinders $\Big\} = 0{\cdot}865\,(37{\cdot}89 - 4)\,\dfrac{3{\cdot}5}{2} = 51{\cdot}3$ lbs.

Then, actual effective mean pressure in low-pressure cylinder $\Big\} = 0{\cdot}865\,(37{\cdot}89 - 4)\tfrac{1}{2} = 14{\cdot}65$ lbs.

And the actual pressure in receiver is then - - - - - - $\Big\} = 76{\cdot}18 - \dfrac{3{\cdot}5}{2}(29{\cdot}3) = 24{\cdot}88$ lbs.

(2) *The three-cylinder receiver compound engine, having two low-pressure cylinders.* Ratio of each low-pressure cylinder to the high-pressure is $\dfrac{R}{2}$.

In this case only one-third of the work is done in each cylinder. Then

$$p'_m - p_r = \frac{R}{2}(p''_m - p_0), \quad - \quad - \quad - \quad - \qquad (1)$$

and as

$$\frac{p'_m - p_r}{R} + (p''_m - p_0) = P_m - p_0.$$

Then

and

$$\left.\begin{aligned} p''_m - p_0 &= \tfrac{2}{3}(P_m - p_0) \\ p'_m - p_r &= \frac{R}{3}(P_m - p_0) \end{aligned}\right\} \quad - \quad - \quad - \qquad (2)$$

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Also the actual values

$$\left.\begin{array}{l} p''_m - p_0 = \frac{2}{3}(P_m - p_0)x \\ p'_m - p_r = \frac{R}{3}(P_m - p_0)x \end{array}\right\} \quad \cdot \quad \cdot \quad \cdot \qquad (3)$$

Example.—To find the mean pressures in a three-cylinder compound engine (having two low-pressure cylinders), using steam of 90 lbs. absolute pressure, the total rate of expansion being seven, the ratio of the combined capacity of the low-pressure cylinder to that of the high-pressure being 3·5, and the back pressure 4 lbs.

$$P_m = 90 \times 0{\cdot}421 = 37{\cdot}89 \text{ lbs.},$$

$$r_1 = \frac{r}{R} = 7 \div 3{\cdot}5 = 2,$$

$$p'_m = 90 \times 0{\cdot}8465 = 76{\cdot}18 \text{ lbs.},$$

$$p_r = 76{\cdot}18 - \frac{3{\cdot}5}{3}(37{\cdot}89 - 4) = 36{\cdot}64 \text{ lbs.},$$

$$r_2 = \frac{p_r r}{p_1} = \frac{36{\cdot}64 \times 7}{90} = 2{\cdot}85,$$

$$p''_m = 36{\cdot}64 \frac{1 + \text{hyp. log. } 2{\cdot}85}{2{\cdot}85} = 26{\cdot}4$$

$$\left.\begin{array}{l}\text{Then the mean effective pressure in the high} \\ \quad\text{pressure cylinder} \;\cdot\;\cdot\;\cdot\;\cdot\;\cdot \end{array}\right\} = 76{\cdot}18 - 36{\cdot}64 = 39{\cdot}54 \text{ lbs.}$$

$$\left.\begin{array}{l}\text{Then the mean effective pressure in the low} \\ \quad\text{pressure cylinder} \;\cdot\;\cdot\;\cdot\;\cdot\;\cdot \end{array}\right\} = 26{\cdot}4 - 4 = 22{\cdot}4 \text{ lbs.}$$

Then

$$P'_m - p_0 = 22{\cdot}4 + \frac{39{\cdot}54}{3{\cdot}5} = 32{\cdot}7 \text{ lbs.}$$

Then

$$x = 32{\cdot}7 \div 33{\cdot}89 = 0{\cdot}965.$$

$$\left.\begin{array}{l}\text{Then actual effective mean} \\ \quad\text{pressure in high-pressure} \\ \quad\text{cylinder} \;\cdot\;\cdot\;\cdot\;\cdot \end{array}\right\} = \frac{3{\cdot}5}{3}(37{\cdot}89 - 4) \times 0{\cdot}965 = 38{\cdot}15 \text{ lbs.}$$

$$\left.\begin{array}{l}\text{Then actual effective mean} \\ \quad\text{pressure in each low-} \\ \quad\text{pressure cylinder} \;\cdot\;\cdot \end{array}\right\} = \frac{2}{3}(37{\cdot}89 - 4) \times 0{\cdot}965 = 21{\cdot}8 \text{ lbs.}$$

(3) *The three-cylinder compound continuous expansion engine, having one high-pressure, one low-pressure, and one medium-pressure cylinder, generally called triple-compound.*

R is the ratio of low-pressure to high-pressure cylinder; R_1 the ratio of low-pressure to mean-pressure cylinder; p' the initial pressure, &c., as before.

p'_m the mean pressure due to expansion, r_1, and pressure p',
p''_m ,, ,, ,, r_2, ,, p'',
p'''_m ,, ,, ,, r_3, ,, p'''.

p'' is the pressure in the receiver between high-pressure and mean-pressure cylinders, p''' that in the receiver between mean-pressure and low-pressure cylinders.

Then effective mean pressure in high-pressure cylinder $= p'_m - p''$,
" " mean-pressure " $= p''_m - p'''$,
" " low-pressure " $= p'''_m - p_0$.

Then, if there is no loss due to drop,

$$p'_m - p'' = R(p'''_m - p^0) \text{ and } p''_m - p''' = R_1(p'''_m - p^0). \quad - \quad (1)$$

But

$$P_m - p^0 = p'''_m - p^0 + \frac{p''_m - p'''}{R_1} + \frac{p'_m - p''}{R}.$$

Therefore

$$\left.\begin{aligned} p'''_m - p^0 &= \frac{P_m - p^0}{3} \\ p''_m - p''' &= \frac{R_1}{3}(P_m - p^0) \\ p'_m - p'' &= \frac{R}{3}(P_m - p^0) \end{aligned}\right\} \quad - \quad - \quad - \quad - \quad - \quad - \quad (2)$$

This is true when there is no loss from "drop," but, as in practice there is generally some loss from this cause, an approximation must be found in a similar way to that for the two-cylinder compound engine.

The cut-off in the high-pressure cylinder will be, as before,

$$\frac{1}{r_1} = \frac{R}{r}.$$

The cut-off in mean-pressure cylinder in order to maintain a pressure, p'', in the receiver between it and the high-pressure cylinder can be found in the same way as before. Since R is the ratio of low-pressure to high-pressure cylinder, and R_1 that of low-pressure to mean-pressure cylinder, $\frac{R}{R_1}$ will be the ratio of the mean-pressure to high-pressure cylinder, then

$$\frac{p''}{r_2} \times \frac{R}{R_1} = \frac{p'}{r_1},$$

and

$$\frac{1}{r_2} = \frac{R_1}{R} \times \frac{p'}{p''} \times \frac{1}{r_1}.$$

Substituting the value of $\frac{1}{r_1}$. Then, $\frac{1}{r_2} = R_1 \frac{p'}{p''r}$.

The cut-off in low-pressure cylinder to maintain a pressure, p''', in the receiver between it and the mean-pressure cylinder,

$$\frac{1}{r_3} = \frac{1}{R_1} \times \frac{p''}{p'''} \times \frac{1}{r_2}.$$

Substituting the value of $\frac{1}{r_2}$. Then, $\frac{1}{r_3} = \frac{p'}{p'''r}$.

But since the terminal pressure in the low-pressure cylinder will be that due to an initial pressure, p', and a rate of expansion, r. Then

$$\frac{p'}{r} = \frac{p'''}{r_3}; \text{ or } \frac{1}{r_3} = \frac{p'}{p'''r}.$$

Therefore,

$$\left.\begin{aligned} &\text{Cut-off in high-pressure cylinder} = \frac{R}{r} \\ &\quad ,, \quad \text{mean-pressure} \quad ,, \quad = R_1 \times \frac{p'}{p''r} \\ &\quad ,, \quad \text{low-pressure} \quad ,, \quad = \frac{p'}{p'''r} \end{aligned}\right\} \quad - \quad - \quad (3)$$

To avoid any lengthy or elaborate calculations, a result sufficiently accurate for practical purposes may be obtained by assuming a value for x, and using it only in the first step of the calculation. This value will vary from 1·0 to 0·9 in well-proportioned engines of this class, when the steam pressure is not less than 120 lbs. absolute, and the rate of expansion not less than 10 times.

Example.—To find the mean pressures in the three-cylinder continuous expansion engine, using steam of 120 lbs. absolute pressure, and expanding 12 times. The ratio of low-pressure to high-pressure cylinder being 6, and of low-pressure to mean-pressure cylinder, 2·5; the back pressure in low-pressure cylinder being 4 lbs.:

Assume $x = 0·9$.

$$\left.\begin{aligned} P_m &= 120 \times 0·2904 = 34·85 \text{ lbs.} \\ p'_m &= 120 \times 0·8465 = 101·58 \text{ lbs.} \end{aligned}\right\} \text{Table ix.}$$

$$p'_m - p'' = \tfrac{6}{3}\,(34·85 - 4) \times 0·9 = 55·53 \text{ lbs.}$$

Therefore,

$$p'' = 101·58 - 55·53 = 46·05 \text{ lbs.}$$

Now,

$$\frac{1}{r_2} = 2·5 \times \frac{120}{46·05 \times 12} = ·544; \text{ or } r_2 = 1·838.$$

Then,

$$p''_m = p''\,\frac{1 + \text{hyp. log. } 1·838}{1·838} = 38 \text{ lbs.}$$

If the work performed in the second cylinder is to equal that done in the first, then

$$p''_m - p''' = \frac{R_1}{R} \times 55·53 = \frac{2·5}{6} \times 55·53 = 23·14 \text{ lbs.}$$

Then,

$$p''' = 38 - 23·14 = 14·86 \text{ lbs.}$$

$$\frac{1}{r_3} = \frac{120}{14·86 \times 12}; \text{ or } r_3 = 1·486.$$

$$p'''_m = p'''\,\frac{1 + \text{hyp. log. } 1·486}{1·486} = 13·96 \text{ lbs.}$$

$$p'''_m - p_0 = 13·96 - 4 = 9·96 \text{ lbs.}$$

Therefore, the mean pressures are 55·53 lbs., 23·14 lbs., and 9·96 lbs.
Referred to the low-pressure cylinder,

$$P'_m - p_0 = 9·96 + \frac{27·14}{2·5} + \frac{55·53}{6} = 28·471,$$

$$P_m - p_0 = 30·85.$$

Therefore,

$$x = \frac{28 \cdot 471}{30 \cdot 85} = 0 \cdot 923.$$

So that if the work is exactly equally divided between the cylinders, then—
Mean pressure in low-pressure cylinder

$$= \frac{P_m - p_0}{3} 0 \cdot 923 = \frac{30 \cdot 85}{3} \times 0 \cdot 923 = 9 \cdot 49 \text{ lbs.}$$

Mean pressure in mean pressure cylinder

$$= \frac{R_1}{3}(P_m - p_0)\, 0 \cdot 923 = \frac{2 \cdot 5}{3} \times 30 \cdot 85 \times 0 \cdot 923 = 23 \cdot 72 \text{ lbs.}$$

Mean pressure in high-pressure cylinder

$$= \frac{R}{3}(P_m - p_0)\, 0 \cdot 923 = \frac{6}{3} \times 30 \cdot 85 \times 0 \cdot 923 = 56 \cdot 94 \text{ lbs.}$$

Actual Mean Pressure in Practice.—In the preceding pages, the mean pressure spoken of is only such as would be obtained from a *perfect* engine in which steam is dry and expanding at a constant temperature, and as such is what may be called the *theoretical mean pressure*. In an actual engine, however carefully designed and manufactured, there will be certain causes of loss of pressure, so that the actual indicator-diagram will show a mean pressure considerably less than that due to the initial pressure and the rate of expansion, allowing that during expansion work has been done.

The following are the principal causes of loss of pressure in the cylinder of a marine engine:—

(1) **Friction in the Stop-valves on the Boiler and Engine, and in the Pipes connecting these.**—If the initial pressure is taken as that in the slide-valve case, of course this particular loss does not affect the indicator-diagram at all. If the stop-valves are opened to the extent of one-quarter of their diameter, and the steam-pipe is fairly straight and short, and of sufficient diameter, so that the flow of steam at any point does not exceed a velocity of 8000 feet per minute: the loss of pressure at the valve-case will be very slight, and not exceed 2½ per cent. If the capacity of the valve-case is nearly equal to that of the cylinder at the cut-off point, the loss will be still less, as the case then acts as a reservoir in which steam is stored between the cut-off and admission periods.

(2) **Friction or Wire-drawing of the Steam during admission and cut-off.**—This is the principal cause of loss of pressure in most marine engines, and is generally due to defective valve-gear, combined with small steam ports and passages. If the opening to steam during admission is small at the most, and the valve closes slowly, large passages and ports are of no avail; and, on the other hand, if the passages are contracted, there will be considerable loss of pressure in the cylinder, however efficient the valve-gearing may be. But the slow and limited motion of the valve itself is the most serious obstacle to the obtaining of good diagrams. The slow opening of the valve causes no loss, as the piston speed is low at that period. A perfect valve should open wide enough to allow the steam to pass at a velocity of 8000 feet per minute, and remain open until cut-off, which should take place quickly; the valve should remain closed until very nearly the end of the stroke, when it should open quickly and wide to *exhaust;* the slow closing to *exhaust*, and slow opening to *lead*, are of no consequence, and cause no

practical loss. The loss of pressure from these causes with engines having common slide-valves, and the ordinary link-motion, is very great; especially is this the case when steam is cut-off early in the stroke, by the main valves being set with very little lead, and having only single ports. As has already been stated, the steam becomes superheated by the friction, and so is a little more efficient during expansion than it would otherwise be.

When cut-off is effected by means of special valve-gear, or by a separate cut-off valve, the pressure at cut-off is very little below that in the valve-casing, and sometimes equal to it; when effected by the ordinary single-ported slide valve and link motion, the pressure at cut-off is sometimes as much as 15 per cent., and seldom less than 7½ per cent. below that in the valve-case.

(3) **Liquefaction during Expansion**, partly due to the cooling action of the walls of the cylinder, is a frequent source of loss of pressure, and this is especially so in expansive engines working with moist steam in unjacketed cylinders; and is observable also, though in a smaller degree, in all compound engines working under similar circumstances.

(4) **Exhausting before the Piston has reached the end of its stroke**, although conducive to the good working of a fast-running engine, will show a loss of pressure in the indicator-diagram. The loss from this cause is, however, more imaginary than real; but it must not be forgotten that the I.H.P. will be less, which is important when the I.H.P. is essential in the contract.

(5) **Compression and Back Pressure due to "Lead,"** also tend to reduce the mean pressure of the diagram when compared with the *theoretical mean pressure*. But these are both essential to the good working of an engine, and (as has been shown in a previous part of this chapter) compression tends to balance the loss due to clearance.

(6) **Friction in the Ports, Passages, and Pipes**, between cylinder and cylinder and condenser, produces a loss of pressure, and, although not large when the velocity through them does not exceed 6000 feet per minute, sometimes amounts to 2 or 3 lbs. in badly designed engines.

(7) **Clearance** has been shown to serve to increase the mean pressure beyond that due to the nominal rate of expansion, and therefore cannot be reckoned as a source of loss, unless the *equivalent* cut-off is taken to obtain the rate of expansion.

It will be seen, then, that the actual mean pressure expected to be obtained from the indicator-diagram of an engine depends very much on the proportion and arrangement of the cylinders and their valves, &c., and in calculating the *expected* mean pressure from the *theoretical mean pressure*, due allowance must be made in each individual case.

If the theoretical mean pressures be calculated by the methods laid down in this chapter, and the necessary corrections made for *clearance* and compression, the *expected* mean pressure may be found by multiplying the results by the factor in the following Table x.

If no correction be made for the effects of clearance and compression, and the engine is in accordance with general modern practice, the clearance and compression being proportionate, then the Theoretical Mean Pressure may be multiplied by 0·96, and the product again multiplied by the proper factor in Table x., the result being the expected mean pressure.

Example.—To find the expected mean pressure in the cylinders of a marine engine using steam of 60 lbs. absolute pressure, the rate of expansion

4, the clearance equal to one-tenth of the cylinder, and the pressure in the condenser 2 lbs., the valve-gearing specially adapted for an early cut-off, and the ports, passages, &c., of ample size; compression commences at ¾ of the stroke. The cylinders are jacketed.

TABLE X.

Particulars of Engine.	Factor.
(1) Expansive engine, special valve-gear, or with a separate cut-off valve, cylinders jacketed, ...	0·94
(2) Expansive engine having large ports, &c., and good ordinary valves, cylinders jacketed, ...	0·9 to 0·92
(3) Expansive engines with the ordinary valves and gear as in general practice, and unjacketed, ...	0·80 to 0·85
(4) Compound engines, with expansion valve to H.P. cylinder; cylinders jacketed, and with large ports, &c., ...	0·9 to 0·92
(5) Compound engines, with ordinary slide valves, cylinders jacketed, and good ports, &c., ...	0·8 to 0·85
(6) Compound engines as in general practice in the merchant service, with early cut-off in both cylinders, without jackets and expansion valves, ...	0·7 to 0·8
(7) Triple-compound engines, with ordinary slide valves, good ports, unjacketed, moderate piston speed, ...	0·65 to 0·75
(8) Fast-running engines of the type and design usually fitted in warships, ...	0·6 to 0·7

The effective rate of expansion is

$$r_1 = 4 \times \frac{1 + 0{\cdot}1}{1 + 4 \times 0{\cdot}1} = 3{\cdot}143.$$

$$p^c = \frac{0{\cdot}25 + 0{\cdot}1}{0{\cdot}1} \times 2 = 7 \text{ lbs.}$$

and the rate of expansion 3·5. Then,

$$p^0{}_m = 7 \times \frac{1 + \text{hyp. log. } 3{\cdot}5}{3{\cdot}5} = 4{\cdot}5 \text{ lbs.}$$

$$p'_m = 60 \times \frac{1 + \text{hyp. log. } 3{\cdot}143}{3{\cdot}143} = 41 \text{ lbs.}$$

$$\text{Expected mean pressure} = 41\,(1 + 0{\cdot}1) - 60 \times 0{\cdot}1 - 2\,(1 + 0{\cdot}5 - 0{\cdot}1) - 4{\cdot}5\,({\cdot}25 + 0{\cdot}1) = 35{\cdot}3 \text{ lbs.}$$

If the effects of clearance and cushioning be neglected, the mean pressure = 60 × 0·5965 – 2, or 33·8 lbs. This is less than the result obtained by the more accurate calculation in this case, because the cushioning is small for so low a back pressure when compared with the clearance.

The mean pressure in practice will be found now by multiplying 35·3 lbs. by 0·94, and is therefore 33·18 lbs.

Example.—To find the expected mean pressure in the cylinders of a compound engine using steam of 100 lbs. absolute pressure, the cut-off in both high-pressure and low-pressure cylinders being at half stroke; the clearance in both cylinders is equal to one-tenth of their net capacity; the pressure in the condenser is 2 lbs.; the cylinders are jacketed, and the ports, &c., of ample size, no expansion valves. Compression commences at ¾ the stroke. Cylinder ratio 4.

Here the effective rate of expansion $= 2\dfrac{1 + 0{\cdot}1}{1 + 2 \times 0{\cdot}1} = 1{\cdot}83.$

The theoretical pressure in the receiver $= \dfrac{100}{1{\cdot}82} \times \dfrac{2}{4} = 27{\cdot}3$ lbs.

The *expected* pressure in receiver $= 27{\cdot}3 \times 0{\cdot}85 = 23{\cdot}2$ lbs.

The steam is compressed in high-pressure cylinder to

$$p^c = \frac{\frac{1}{4} + \frac{1}{10}}{\frac{1}{10}} \times 23{\cdot}2 = 81{\cdot}2 \text{ lbs.}$$

The rate of compression $= \dfrac{\frac{1}{4} + \frac{1}{10}}{\frac{1}{10}} = 3{\cdot}5.$

The mean pressure due to a rate of expansion 1·83, and an initial pressure of 100 lbs.

$$= 100\,\frac{1 + \text{hyp. log. } 1{\cdot}83}{1{\cdot}83} = 87 \text{ lbs.}$$

The mean pressure due to a rate of expansion 3·5, and an initial pressure of 81·2 lbs.

$$= 81{\cdot}2\,\frac{1 + \text{hyp. log. } 3{\cdot}5}{3{\cdot}5} = 52 \text{ lbs.}$$

The theoretical mean pressure in high-pressure cylinder

$$= 87\,(1 + 0{\cdot}1) - 100 \times 0{\cdot}1 - 23{\cdot}2\,(1 - 0{\cdot}5 - 0{\cdot}1) - 52\,(0{\cdot}25 + 0{\cdot}1)$$
$$= 58{\cdot}22 \text{ lbs.}$$

The *expected mean pressure* in high-pressure cylinder

$$= 58{\cdot}22 \times 0{\cdot}85 = 49{\cdot}5 \text{ lbs.}$$

The mean pressure due to a rate of expansion 1·83, and an initial pressure of 23·2 lbs.

$$= 23{\cdot}2\,\frac{1 + \text{hyp. log. } 1{\cdot}83}{1{\cdot}83} = 20{\cdot}2 \text{ lbs.}$$

The mean pressure due to a rate of expansion 3·5, and an initial pressure of 7 lbs.

$$= 4{\cdot}5 \text{ lbs.}$$

Then theoretical mean pressure in low-pressure cylinder

$$= 20{\cdot}2\,(1 + 0{\cdot}1) - 23{\cdot}2 \times 0{\cdot}1 - 2\,(1 - 0{\cdot}5 - 0{\cdot}1) - 4{\cdot}5\,(0{\cdot}25 + 0{\cdot}1)$$
$$= 17{\cdot}5 \text{ lbs.}$$

And the *expected mean pressure* in low-pressure cylinder

$$= 17{\cdot}5 \times 0{\cdot}85 = 14{\cdot}87 \text{ lbs.}$$

Example.—To find the expected mean pressure in a compound engine as fitted in ordinary merchant steamers; the cylinders are unjacketed, the boiler pressure 80 lbs. (95 lbs. absolute); the cylinder ratio is 3·5, and the cut-off, effected by common slide-valves, is at half-stroke in the high-pressure cylinder, and 0·6 the stroke in low-pressure cylinder. The clearance in both cylinders is one-twelfth the cylinder capacity. Compression takes place in the high-pressure cylinder when the piston is 0·2 of its stroke from the end, and in the low-pressure cylinder at 0·3.

Efficiency in this case taken at 0·75.

The effective rate of expansion in high-pressure cylinder

$$= 2\frac{1 + \frac{1}{12}}{1 + \frac{2}{12}} = 1{\cdot}86.$$

The theoretical pressure in the receiver

$$= \frac{95}{1{\cdot}86} \times \frac{1}{3{\cdot}5 \times 0{\cdot}6} = 24{\cdot}3 \text{ lbs}$$

The expected pressure in the receiver

$$= 24{\cdot}3 \times 0{\cdot}75 = 18{\cdot}23.$$

The rate of compression in the high-pressure cylinder

$$= \frac{0{\cdot}2 + 0{\cdot}083}{0{\cdot}083} = 3{\cdot}4,$$

and the steam is compressed to 18·23 × 3·4, or 62 lbs. The mean pressure due to a rate of expansion of 1·86, and an initial pressure of 95 lbs.

$$= 95\,\frac{1 + \text{hyp. log. } 1{\cdot}86}{1{\cdot}86} = 80 \text{ lbs.}$$

The mean pressure due to a rate of expansion 3·4, and an initial pressure of 62 lbs.

$$= 62\,\frac{1 + \text{hyp. log. } 3{\cdot}4}{3{\cdot}4} = 40 \text{ lbs.}$$

Then theoretical mean pressure in high-pressure cylinder

$$= 80(1 + \tfrac{1}{12}) - 95 \times \tfrac{1}{12} - 18{\cdot}23(1 - 0{\cdot}4 - \tfrac{1}{12}) - 40(0{\cdot}2 + \tfrac{1}{12})$$
$$= 58 \text{ lbs.}$$

And the expected mean pressure in high-pressure cylinder

$$= 58 \times 0{\cdot}75 = 43{\cdot}5 \text{ lbs.}$$

The back pressure in the condenser is 2 lbs.

The effective rate of expansion in low-pressure cylinder

$$= \frac{1}{0{\cdot}6} \times \frac{1 + \dfrac{1}{12}}{1 + \dfrac{1}{0{\cdot}6}\,\dfrac{1}{12}} = 1{\cdot}58.$$

The rate of compression in low-pressure cylinder

$$= \frac{0{\cdot}3 + 0{\cdot}083}{0{\cdot}083} = 4{\cdot}6.$$

Steam is compressed in low-pressure cylinder to 4·6 × 2, or 9·2 lbs.

The mean pressure due to a rate of expansion 1·58, and an initial pressure of 18·23 lbs.

$$= 18{\cdot}23 \times \frac{1 + \text{hyp. log. } 1{\cdot}58}{1{\cdot}58} = 16{\cdot}8 \text{ lbs.}$$

The mean pressure due to a rate of expansion 4·6, and an initial pressure 9·2 lbs.

$$= 9{\cdot}2 \times \frac{1 + \text{hyp. log. } 4{\cdot}6}{4{\cdot}6} = 5 \text{ lbs.}$$

Then theoretical mean pressure in low-pressure cylinder

$$= 16{\cdot}8\,(1 + \tfrac{1}{12}) - 18{\cdot}23 \times \tfrac{1}{12} - 2\,(1 - 0{\cdot}6 - \tfrac{1}{12}) - 5\,(0{\cdot}3 + \tfrac{1}{12})$$
$$= 14{\cdot}13 \text{ lbs.}$$

And the *expected mean pressure* in low-pressure cylinder

$$= 14{\cdot}13 \times 0{\cdot}75 = 10{\cdot}6 \text{ lbs.}$$

Practical Method of Calculating the Expected Mean Pressure.—An approximate value for the expected mean pressure in a compound engine may be found by first calculating the theoretical mean pressure due to the total rate of expansion, subtracting from it the back pressure in condenser, and dividing by 2 for a two-cylinder engine, by 3 for a triple, and by 4 for a quadruple engine. The result is the mean pressure in the low-pressure cylinder, when there is no loss from "drop." Multiply this by the factor in Table x., p. 123, and again multiply the product by 0·8, and the *result is the expected mean pressure when the work is equally divided between the two cylinders.*

Example.—To find the mean pressures expected in the cylinders of a compound engine, using steam of 90 lbs. absolute pressure, and expanding it six times; the ports being of ample size, and the cylinders jacketed; the cut-off in high-pressure cylinder effected by an expansion-valve, and the pressure in the condenser is 3.

The mean pressure due to a rate of expansion of 6, and initial pressure of 90 lbs. - - - - - - - = 90 × ·4653 = 41·8 lbs.
The effective mean pressure - - - - = 41·8 − 3 = 38·8 lbs.
The effective mean pressure in L.P. cylinder = 19·4 lbs.
The expected mean pressure in L.P. cylinder = 19·4 × 0·9 × 0·8 = 13·9 lbs.

If the ratio of the cylinder is 3·5, then

The expected mean pressure in H.P. cylinder = 13·9 × 3·5 = 48·7 lbs.

Graphic Method.—It has been assumed in this chapter that steam expands in accordance with Boyle and Marriott's law—viz., $p\,v$ = *constant*—on account of its convenient form and easy calculation. As a matter of fact, the results obtained in this way are sufficiently accurate for practical purposes, although, of course, they are not scientifically accurate, for steam does not expand in practice at a constant temperature as is assumed in Boyle's law, and, moreover, when expanding in practice, it is doing work, and therefore giving up some of its heat with a corresponding reduction in pressure.

The simpler and more easily-worked methods of obtaining the division of power in compound engines are graphic—*i.e.*, by means of geometrical diagrams based on the same physical facts as before.

Draw a straight line A C (fig. 27) to represent the capacity of the low-pressure cylinder, including its clearance, which latter is represented by A B.

Draw an ordinate, A K, representing the absolute pressure of the steam.

Take a point, D, on A C, so that $\frac{A\,C}{A\,D}$ is the total rate of expansion required in the system.

Then draw the expansion curve, D Q′ C′.

Take a point, F′, in K D, so that K F′ represents the clearance of the high-pressure cylinder.

Draw F′ F parallel to A K.

Now, K D′ represents the capacity of the high-pressure cylinder, including its clearance at the point of cut-off in that cylinder.

Then, take a point, L, in A C, so that $\frac{F\,D}{F\,L}$ = rate of cut-off in the high-pressure cylinder, and draw L L′ parallel to A K.

A L will therefore represent the capacity of the high-pressure cylinder, including its clearance A F.

Now, take point N on A C, so that A N represents the clearance in the medium-pressure cylinder.

Take a point, V, in A C, and a point, Q, in A C, so that A Q represents the volume of the medium-pressure cylinder, including its clearance, A N, and $\frac{\mathrm{N\,V}}{\mathrm{N\,Q}}$ = the rate of cut-off in the medium-pressure cylinder.

Draw V V′ and Q Q′ parallel to A K.

Take a point, S, in A C, so that $\frac{\mathrm{B\,S}}{\mathrm{B\,C}}$ = the rate of cut-off in low-pressure cylinder.

Draw S S′ parallel to A K, and S′ R parallel to A C.

Then F′ D′ L′ L″ F″ is the theoretical high-pressure diagram.

N′ V′ Q′ Q″ N″ is the medium-pressure diagram, and B′ S′ C′ C B the low-pressure diagram.

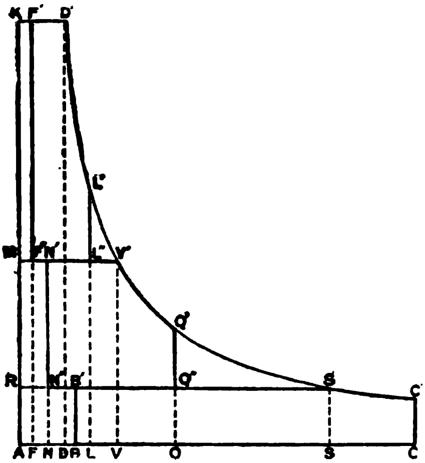

Fig. 27.—Theoretical Diagram for Triple-Expansion.

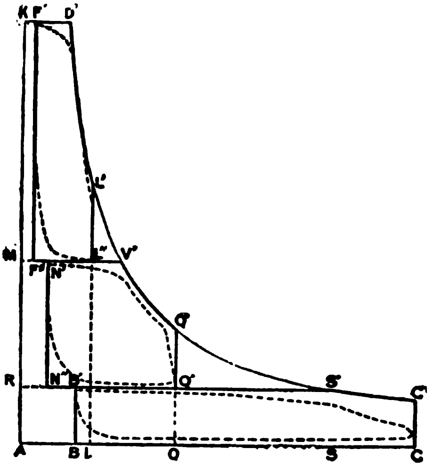

Fig. 28.—Diagrams as in Practice, from Theoretical Expansion Diagram.

In actual practice the indicator diagrams differ from the theoretical ones, for reasons already given. They may, however, be inscribed within the theoretical diagram of the mean pressure, as shown in fig. 28, which has been drawn in accordance with the method prescribed above, so that F′ D′ F is the high-pressure diagram, V Q N the medium, and S C′ C B the low. In practice, the area of the actual diagram of the high pressure is 77 to 80 per cent. of the theoretical. The actual medium-pressure diagram is 70 to 73 per cent. of the theoretical, and the low pressure 55 to 60 per cent.

In fast-running engines the percentage is, of course, less than that obtained from the diagrams of slow-moving engines.

If it is found by this means that the power is not sufficiently evenly divided, the cuts-off in the medium-pressure and low-pressure cylinders must be modified. For example, if the power in the low-pressure cylinder is too small, the cut-off point, S, must be moved nearer to A, so that the figure, B′ S′ C′ C B, is enlarged at the expense of the medium-pressure diagram, N′ V′ Q′ Q″ N″

TABLE XI.—Results of Trials of Engines of Naval Ships (Twin Screw).

Name.	Nature of Trials.	Cylinders.					Engine Room.									One Engine					Coal per I.H.P.	Nominal Rate of Expansion.
		Diameters.				Stroke. Inches.	Steam. Lbs.	Vacuum. Inches.	Revolutions.	Cut-Off. H.P.	Mean Pressures.					Indicated Horse Power.						
		H.P.	M.P.	L.P.	L.P.						H.P.	M.P.	L.P.	L.P.	Referred to L.P. Cyl.	H.P.	M.P.	L.P.	L.P.	Total.		
Good Hope,	Full S. Red. S.	43½	71	81½	81½	48	278 268	27 26·5	126 110·7	0·78 0·68	105 95·8	44·6 37·9	16·9 11·9	17·9 13·4	49·3 40·7	4,764 3,819	5,399 4,029	2,698 1,606	2,849 1,864	15,710 11,378	1·70 1·75	9·60 ..
Argonaut,	Full S. Red. S.	34	55½	64	64	48	236 242	25·5 27	129·8 115·7	0·71 0·60	96·5 89	32·2 27·9	19·2 17·6	23·1 21·5	46·1 42·7	2,704 2,266	2,406 1,900	1,908 1,595	2,295 1,950	9,313 7,711	1·66 1·73	9·99 11·82
Formidable,	Full S. Red. S.	31½	51½	84	..	51	245 240	25·5 26·5	110·2 101	0·70 ..	115 95·2	44 34·4	17·9 14·3	.. ..	50·5 40·7	2,543 1,934	2,601 1,868	2,815 2,066	.. ..	7,959 5,868	1·80 2·00	10·10 ..
St. George,	Full S.	40	59	88	..	51	154	27·8	100·1	0·67	45·2	24·3	13·1	..	33·4	1,465	1,713	2,055	..	5,233	1·62	7·22
Friesland,	Full S.	33	49	74	..	39	200	24·8	143	..	63	36·5	14·74	..	43·2	1,549	2,001	1,851	..	5,401	..	..
Arrogant,	Full S. Red. S.	26	42	68	..	39	245 235	26 27	143 128	0·70 0·514	112·5 86·5	47·1 35·3	20·5 15·3	.. ..	52·3 41·2	1,685 1,159	1,841 1,233	2,100 1,449	.. ..	5,406 3,841	.. ..	9·77 13·50
Antrim,	Full S. Red. S.	41½	65½	73½	73½	42	185 161	25·0 25·5	142·6 87·7	.. ..	82·6 33·12	32·5 12·05	16·7 3·77	16·6 3·34	42·7 14·7	3,333 960	3,315 756	2,143 299	2,186 303	10,977 2,318	2·26 2·06	.. ..
Pearl,	Force D. Natl. D.	30½	45	68	..	33	145 149	24·8 27	158·3 143·1	0·85 ..	56·9 46·3	30 23·7	13·1 10·9	.. ..	37·1 30·6	1,098 806	1,260 901	1,253 942	.. ..	3,611 2,649	.. ..	5·85 ..
Perseus,	Full S. Red. S.	20½	33	54	..	27	266 251	24·2 25·1	214·3 197	0·75 ..	.. ..	.. ..	.. ..	.. ..	52·5 42·9	992 707	1,263 987	1,268 942	.. ..	3,523 2,636	.. 2·27	9·83 ..
Forward,	Full S. Red. S.	31½	50½	57	57	30	227 50	25·3 24·0	210·6 80·3	0·58 0·44	109 17·1	34·5 5·7	14·85 1·87	14·88 1·76	44·5 6·6	2,727 162	2,224 139	1,218 58	1,217 57	7,386 414	2·71 2·66	11·4 15·0
Topaze,	Full S. Red. S.	24½	38½	42½	42½	24	271 198	24·0 25·0	245·5 150·3	.. ..	111·0 34·63	47·2 15·15	21·5 10·36	23·7 9·75	59·5 22·9	1,534 292	1,637 321	897 265	946 249	5,014 1,127	2·65 2·06	.. ..
Dove,	Force D.	19½	30	34	34	18	206	25	385·4	0·75	77·3	45·5	17·7	17·2	47·4	808	1,127	563	547	3,045	2·38	8·09
Snapper,	Force D.	19½	28½	43	..	18	180	25	370·6	0·58	64·5	28·7	14·6	..	39·7	635	592	715	..	1,942	..	8·62
Speedy,	Force D. Natl. D.	22	34	51	..	21	198 189	27·5 27·5	247 209·7	.. ..	75·6 61·5	33·4 26·3	15·6 12·1	.. ..	44·5 35·4	749 522	795 533	847 551	.. ..	2,391 1,606	.. ..	.. ..

TABLE XI*a*.—RESULTS OF TRIALS OF ENGINES OF MERCHANT SHIPS.

NAME.	Twin or Single Screw.	CYLINDERS.					ENGINE ROOM.									ONE ENGINE.					Nominal Rate of Expansion.
		Diameters.				Stroke. Inches.	Steam. Lbs.	Vacuum. Inches.	Revolutions.	Cut-Off. H.P.	Mean Pressures.					Indicated Horse Power.					
		H.P.	M.P.	3rd Cylr.	L.P.						H.P.	M.P.	3rd Cylr.	L.P.	Referred to L.P.	H.P.	M.P.	3rd Cylr.	L.P.	Total.	
Auguste Victoria,	T.S.	41	67	106	..	63	149·3	24	78	0·78	62·6	27·8	..	11·16	32·8	2,214	2,470	..	2,494	7,178	8·58
Kherson, - - -	T.S.	36	57	92	..	54	155	27½	90·6	..	82·9	36·8	..	14·2	41·0	2,085	2,320	..	2,335	6,740	..
Tamaulipa, - - -	S.S.	40	64	92	..	60	140	23	77	0·735	55·1	21·4	..	11·45	32·2	1,615	1,606	..	1,775	4,996	7·20
Saxonia, - - - -	T.S.	29	41½	59	84	54	192	25·3	78	..	78·1	41	20·9	12·08	39·0	985	1,158	1,202	1,420	4,765	..
Liguria, - - - -	S.S.	33	54	88	..	60	183	25	72	..	77·5	30	..	12	34·2	1,444	1,499	..	1,592	4,535	..
Cazengo, - - - -	S.S.	32	48	80	..	48	145	..	71	0·656	68·1	32·0	..	11·0	33·6	943	994	..	947	2,884	9·50
Martello, - - -	S.S.	31	50	82	..	57	142	24	59	0·60	56·2	21·1	..	10	26·0	720	704	..	897	2,321	11·67
Othello, - - - -	S.S.	24½	40	70	..	48	200	28	71	0·59	88·5	36·8	..	13·6	36·7	783	794	..	903	2,480	15·25
Ariosto, - - - -	S.S.	30	46	75	..	45	148	..	79	0·68	57·4	33·4	..	11·05	32·8	728	996	..	876	2,600	9·92
Eldorado, - - -	S.S.	28	43	70	..	39	154	28	88	0·60	57·6	31·2	..	10·7	31·6	616	785	..	710	2,112	10·40
Amsterdam, - -	T.S.	26	39½	61	..	36	155	26½	135·6	0·71	62·1	28·4	..	14·5	37·8	812	857	..	1,048	2,717	7·75
P. George, - - -	T.S.	26	40	45	45	30	177	27	177	0·69	57·5	30·3	15·4	15·3	36·9	818	1,019	682	630	3,149	8·89
Victoria, - - - -	S.S.	23½	38	57	..	33	150	24½	105·6	0·75	56·7	31·1	..	12·8	34·1	433	526	..	573	1,532	7·87
Ohio, - - - - -	S.S.	22½	32	45	64	42	200	27½	67·5	0·64	63·5	28·5	20·4	11	39·0	362	328	464	507	1,661	12·66
P. Edward, - - -	T.S.	19	30	48	..	24	180	25	192·3	0·68	81	33·5	..	13·7	39·6	536	554	..	577	1,667	9·37
Castro, - - - -	S.S.	19	33	56	..	36	200	25	91·5	0·59	87·8	33·5	..	10·9	32·5	417	470	..	445	1,333	14·69
Iona, - - - - -	S.S.	22	34	57	..	39	150	27·3	61·5	0·25	47	20	..	7·2	21·7	208	217	..	220	645	26·88
Colchester, - - -	T.S.	14½	23	27	27	33	180	26·5	145	0·75	88	38·0	18·4	18·2	44·6	331	382	255	252	1,220	9·33
Northern, - - -	S.S.	26	..	..	56	60	115	27·5	57·5	..	66·9	..	..	14·5	30·0	619	..	..	622	1,241	7·78
Bolama, - - - -	S.S.	18	27	48	..	24	181	26·7	89	0·56	75	41·3	..	10·25	38·1	231	286	..	225	742	12·71
Righi, - - - - -	S.S.	17	25	35	50	27	184	25	72	0·57	61·8	32·6	20·1	7·8	32·7	136	157	189	151	633	15·18
Molopo, - - - -	S.S.	13	22	36	..	24	200	23·6	112	0·60	96·7	38·7	..	13·1	40·0	174	199	..	181	554	12·75
Polo, - - - - -	S.S.	12½	18½	27	39	24	200	25	98	0·62	75	34·3	15·3	11·85	34·8	113	110	104	168	495	15·05
Gertie, - - - -	S.S.	12	20	32	..	23	200	22	136	0·58	74·5	38·8	..	15·5	41·0	133	192	..	197	522	12·26

TABLE XI*b*.—RESULTS OF TRIALS OF MARINE ENGINES.

GENERAL DESCRIPTION OF ENGINES.	CYLINDERS.		N.H.P.	Boiler Pressure.	Vacuum.	Revolutions.	Rate of Total Expansion.	CYLINDER MEAN PRESSURE.		I.H.P.		
	Diameter.	Stroke.						H.P.	L.P.	H.P.	L.P.	Total.
	ins.	ins.		lbs.	ins.			lbs.	lbs.			
Screw Vertical Compound, .	1 of 62 2 of 90	66	1000	86	27	55	...	44	16·5 16·7	2433	1945 1928	6306
Do. do. do., .	43 and 86	60	420	80	27	57·4	7·0	47·4	11·2	1198	1131	2329
Do. do. do., .	46 and 87	57	400	76	28	57	5·84	41·2	12·7	1124	1238	2362
Do. do. do., .	42 and 84	60	400	88	26¾	57	7·2	51·0	11·7	1220	1127	2347
Do. do. do., .	52 and 96	48	400	63	26½	60	...	35·7	10·3	1103	1085	2188
Do. do. do., .	34 and 68	63	275	78	25½	57	6·8	45·8	9·75	754	642	1396
Do. do. do., .	38 and 76	48	270	79	26½	62	6·62	46·4	11·2	790	763	1553
Do. do. do., .	40 and 74	48	250	80	27	63	5·67	41·1	12·26	789	805	1594
Do. do. do., .	32 and 64	63	250	80	25	53	6·9	49·7	11·67	674	633	1307
Do. do. do , .	36 and 72	48	230	72	27¼	60	6·0	45·2	10·3	669	610	1279
Do. do. do., .	35 and 69	39	180	82	26¾	70	6·6	46·4	11·5	614	593	1207
Do. do. do., .	33¾ and 64½	42	180	70	28	58	5·55	40·3	11·3	442	454	896
Do. do. do., .	32 and 62	36	150	66	25	69	6·43	45·4	10·37	458	392	851
Do. do. do., .	20½, 33, 58	36	140	147	25¼	62	14·4	61·6 24·6	7·1	229 232	197	664
Do. do. do., .	25 and 50	45	130	80	27	71	7·5	46·2	12·5	366	397	763
Do. do. do., .	30 and 55	36	125	84	24¼	74·5	6·96	45·1	11·9	430	383	813
Do. do. do., .	27 and 53	33	110	80	24	84	7·49	46·1	11·75	369	361	731
Do. do. do., .	27 and 50	30	100	73	23½	68	5·42	43·6	13·36	260	270	530
Do. do. do., .	24 and 45	33	90	84	26	82	6·43	45·1	13·0	278	282	560
Do. do. do., .	21 and 40	27	60	77	25¾	98	6·15	45·1	11·37	209	191	400
Do. do. do., .	17 and 32	18	35	72	25	120	6·4	40·7	9·8	100	86	186
Screw Horizontal Compound, .	60 and 99	39	400	59½	26	85	...	28·5	9·15	1347	1179	2526
Do. do. do., .	57 and 90	33	356	60	25¼	95·8	...	25·7	10·65	1051	1088	2139
Do. do. do., .	31 and 48	18	60	60	26	137	...	25·4	12·42	240	279	519

CHAPTER VII.

PISTON SPEED, STROKE OF PISTON, REVOLUTIONS, SIZE OF CYLINDER, CYLINDER FITTINGS, ETC., ETC.

The Indicated Horse-Power depends on the area of piston, speed of piston, and mean pressure exerted by the steam on the piston; the two latter are variable, which *may* be fixed in an arbitrary way within certain limits, but which must generally depend on the particular circumstances of each individual engine. It has been shown in the last chapter how the mean pressure may be calculated for a proposed engine; but it is necessary to know beforehand the speed of piston, before any decision can be made for the size of the pistons of an engine to develop a certain horse-power.

Piston Speed.—The speed of piston depends on the length of stroke and number of revolutions, and the *mean* velocity is equal to twice the length of stroke multiplied by the number of revolutions per minute.

Experience has shown that heavy marine pistons may be run safely at a velocity of 900 feet per minute, and, in some instances, the pistons of some large vertical engines in first-class cruisers have been run at a velocity of even 1000 feet; while higher speeds still have been attained in torpedo-boat destroyers, whose pistons move, when run at express speed, at a velocity over 1200 feet per minute. Although there is no difficulty in causing a piston to move at even higher speeds than these, it is doubtful if there would be any advantage in doing so, and the risk of causing serious damage to the cylinders, and precipitating a break-down without any warning, is very great. There is no doubt that a well-fitted piston, moving in a smooth and true cylinder at a speed of 1000 feet per minute, will work well so long as the rubbing surfaces receive a steady lubrication from the moisture of the steam or the oil injected, and there is not the slightest fear of danger under these circumstances; but if a little priming occurs, and the scum carried into the cylinders causes abrasion of the rubbing surfaces, an immense amount of mischief may be caused in a few seconds. Moreover, when the cylinders wear a little out of shape from one cause or other, so that the packing-rings will have lateral motion, the danger increases with the velocity of the piston. In the Navy engines supplied with steam from water tube-boilers have no internal *oil* lubrication beyond what passes in on the rod surfaces; so that in destroyers, whose pistons are moving at a speed of 20 feet per second, there is only the moisture of the steam to lubricate them.

Although the revolutions of a screw engine may be, within certain limits, as few or as many as the designer chooses, experience or prejudice has fixed very closely in practice the limits beyond which it is not considered expedient to go. In the days of the geared engine, the screw revolved three or four times to once of the engine, and no objection was raised to the small screw and the high number of revolutions; later such a thing was deemed very objectionable—on the ground of excessive speed of piston by some engineers, and excessive friction in journals by others. And also the slow-moving engine was quoted as a proof of the economy of slow piston speed and small friction.

The fine lines of the older steamships admitted of the small screw, which was the accompaniment of the engine, by necessity geared. Bluff ships, as now built for mercantile purposes, require a much larger screw for the same power of engine and dimensions of hull than formerly obtained; and it is not to the *slowness* of the pistons that they owe their economy, but rather to the *small number* of strokes per minute made by them in turning the large screw.

An engine requires a certain power to be expended in moving it through *one revolution* to overcome internal resistances; if the number of revolutions is 80 per minute, this power will be double that at 40, and, roughly, will vary directly with the revolutions. But the resistance of the propeller, caused by friction of the water on the surface of the blades, will increase roughly as the square of the revolutions, so that the *power* expended to overcome this resistance at 80 revolutions is eight times that required at 40 revolutions. If now the screw can be so altered with respect to pitch that, at 40 revolutions, the same speed of ship is obtained as at 80 revolutions, the indicated horse-power will be found to be considerably less; and although the *coal consumed per I.H.P.* will not be less, and may possibly be more than before, the consumption per day will be considerably less. Now, although this economy is co-existent with decreased piston speed, it is not due to it.

The object of a high rate of piston velocity is to decrease the piston area, and that generally for the sake of reducing the size of the engine. But an increased velocity may be obtained either by increasing the stroke of piston, or by increasing the number of revolutions; if the former method is adopted, there will be no decrease in the size of engine; but, on the contrary, an increase in space occupied and in the weight. If a high piston speed is obtained by a high number of revolutions, a smaller cylinder will suffice for a certain indicated horse-power than if the same piston speed were obtained by length of stroke alone. In other words, engines which are required to develop a certain power in a minute will vary in size of cylinder inversely as the number of revolutions per minute, all other things remaining constant; and if the cylinders are of the same diameter, the stroke will vary inversely as the number of revolutions.

The piston speed of many engines is governed entirely by circumstances beyond the immediate control or will of the designer. An example of this is the case of the paddle-wheel engine with vertical oscillating cylinders. If the position of the shaft is determined by the structural arrangements of the hull, as is often the case, then the diameter of the wheel is fixed, and the speed of ship fixes the number of revolutions to be made by the wheel; the length of stroke of piston is limited by the distance from the centre of the shaft to the floors or keelson of the ship. Further, if the engineer is free to decide the position of the shaft, any attempt to increase the piston speed by placing the shafting high is frustrated by the fact that, the higher the shaft the larger will be the wheel, and consequently the fewer the revolutions. If the engine is inclined, then the designer may fix the diameter of the wheel to suit the revolutions which he deems most advisable, or he may fix the position of shaft to suit the ship's structure, and still be free to choose the stroke of piston.

Again, the horizontal engine must be designed so as to accommodate itself to the space allotted to it in the ship, which means that only a limited length of stroke is permissible. The revolutions, however, in this case may be varied considerably; but there is, after all, a limit to the number; beyond this limit any increase will result in very little gain in speed, and a very certain loss of efficiency. If the screw is of comparatively small diameter, owing to the shallow draught of the ship, a higher number of revolutions than usual is absolutely necessary to project a sufficient mass of water back to propel the ship forward with the necessary velocity; and it is the medium number, or that number at which the engine can be run without loss of efficiency so as to obtain the maximum speed of ship that is so difficult to decide, and which can only be determined with any degree of certainty by experiment.

The one great feature which places the vertical engine so much above all the other forms of screw engine, as an economic and good working machine, is its superior length of stroke. Power for power, the vertical engine always has exceeded the horizontal in this respect; and although in the practice of the past there was no very great difference in this matter between the two types, the tendency is now to make the stroke as long as is possible or convenient in the engines of all ships, and to remain as before in the horizontal engines of warships.

The advantages of the long stroke are due to the corresponding decrease in piston area. Two engines of the same power, and working at the same number of revolutions, must have the same volume of cylinder; or, to speak more correctly, the pistons must sweep out the same volume if their efficiency is the same. The crank-shafts will be of the same diameter, and the crank-pins, also, practically of the same dimensions. Now the one with the long stroke will have smaller pistons than the other, consequently the total pressure on the pistons will be smaller—and, in fact, is inversely proportional to the stroke; consequently, the pressure on the guides, crank-pins, and journals will vary in the same way, and the friction on them correspond also. The lateral pressure of the piston packing rings will vary with the diameter, so that any reduction in diameter will produce a corresponding reduction in the friction.

But, perhaps, so far as economy in working is concerned, there is no more important consideration than the reduction in clearance space effected in low-pressure cylinder by the reduction in piston area. The steam ports will be nearly the same, whether the engine be long or short stroke; but the space between the piston and cylinder-ends is very considerably reduced, and will vary inversely as the length of stroke, because the distance of piston from the cylinder-ends is constant.

Revolutions.—Although there is a very considerable range for choice of number of revolutions of the engines of most merchant steamers, there are certain well-defined limits beyond which very few practical engineers go.

Taking the nominal horse-power to designate the engines, and adopting the following rule to calculate it, D_h being the diameter of the high-pressure cylinder, D_m that of the medium, and D_l that of the low-pressure cylinder, and S the length of stroke, all in inches,

$$\text{N.H.P.} = \frac{D_h^2 + D_m^2 + D_l^2}{100} \sqrt[3]{S}.$$

The following is considered good practice for the engines of ordinary merchant steamers; express cross-channel steamers exceed these considerably:—

TABLE XII.

N.H.P.	Revolutions.	N.H.P.	Revolutions.	N.H.P.	Revolutions.
20	180 to 200	70	120 to 140	140	100 to 115
30	160 to 180	80	115 to 135	180	95 to 110
40	150 to 170	90	110 to 130	200	95 to 105
50	140 to 160	100	105 to 125	250	90 to 100
60	130 to 150	120	100 to 120	upwards	80 to 95

Very few screw engines are now worked below 75 revolutions per minute when in good condition; and it is at this speed that most of the engines of the large mail steamers are kept running on the voyage so long as the weather permits. The engines of warships, for two very good reasons, work at much higher speeds. Their machinery must be light, and go into a small space, so that it is necessary to make an engine of certain dimensions to suit these conditions, and cause it to develop the requisite horse-power by running at a higher number of revolutions. The speed of a warship is much higher in proportion to its size than is that of the merchant ship, while the draught of water is no more, and often less. For these reasons the screw of the warship is small for the power to be developed, so that even if large engines were admissible to drive the screw, they would be of small advantage, as they would have to move at a high rate. It will be seen, then, that small fast-running engines are a necessity, and especially is this so with modern warships, whether armoured or unarmoured. The latter must be as fine as possible, and every ton of weight saved to obtain the very high speeds which their service demands; the former demands every sacrifice to save weight in machinery, for the sake of adding it to the armour and armament.

Since a warship has so seldom to steam at full speed, and when she does, it is only for a short period, the short-stroke fast-running engine is not so very objectionable, and rigid economy is quite a secondary consideration in war questions.

The Admiralty have, for many years, given up using N.H.P., and have adopted I.H.P. to designate the size of their engines. The following table gives the number of revolutions at which naval engines are run on their trial trips:—

TABLE XII*a*.

Ship.	I.H.P.	Revolutions per Minute.	
		Reciprocators.	Turbines.
Torpedo Boats, ...	500-2000	460-380	1200
Sloops, ...	1,400	200	...
Destroyers, ...	4,000-6,000	370-400	1200-950
Cruiser, 3rd Class, ...	10,000-14,000	245-	490-450
Scout Class, ...	15,000	210	700
Cruiser, 2nd Class, ...	15,000-20,000	145	...
Do., 1st do., ...	22,000-30,000	145-126	300
Battleship, ...	15,000-23,000	120-110	300

The stroke of horizontal engines varied from 18 inches of the gunboat to 54 inches of the large armour-clad, and the vertical engines of the Navy vary from 18 inches in the "destroyers" to 51 inches in the first-class cruisers and battleships.

Length of Stroke.—For very many years there existed a standard scale for the stroke of the vertical engine of the mercantile marine, and although there was no written law which guided engineers in the choice of this important dimension, it was so well known that only *the diameter* of the cylinders was mentioned in speaking of the size of engine, and in most of the rules for nominal horse-power used by manufacturing engineers in their dealings with shipowners, no direct allowance was made for length of stroke. With the walls of the cylinders only jacketed the best relation of diameter to stroke was as 1 to 1·5. Even at the present time, in some districts an engine is called a certain N.H.P. whatever the stroke may be. There is still some semblance of a standard, although competition and differences of opinion are steadily sweeping it away; so long as the expression N.H.P. exists, so long there must be some standard stroke for every power of engine; for the rule for N.H.P., which includes the factor $\sqrt[3]{S}$, is not sufficient by itself, as it is manifestly absurd to suppose that there is only so slight a difference in value as would be given by this rule between two engines whose cylinders are of the same diameter, while the stroke of one is double that of the other.

The following Table gives the stroke corresponding to the different powers; the one column giving the standard, and the other the stroke as existing in ordinary everyday practice:—

TABLE XIII.

N.H.P.	Standard Stroke.	Stroke, as in common practice.	N.H.P.	Standard Stroke.	Stroke, as in common practice.
20	12 ins.	15 ins. to 18 ins.	140	33 ins.	33 ins. to 42 ins.
30	15 ins.	18 ins. to 21 ins.	160	33 ins.	33 ins. to 42 ins.
40	18 ins.	20 ins. to 24 ins.	180	36 ins.	36 ins. to 45 ins.
50	21 ins.	24 ins. to 30 ins.	200	36 ins.	36 ins. to 48 ins.
60	24 ins.	24 ins. to 27 ins.	250	39 ins.	39 ins. to 54 ins.
80	27 ins.	27 ins. to 30 ins.	300	42 ins.	42 ins. to 54 ins.
100	30 ins.	30 ins. to 36 ins.	400	45 ins.	45 ins. to 60 ins.
120	30 ins.	30 ins. to 39 ins.	500	48 ins.	48 ins. to 66 ins.

Some engineers go beyond the strokes given in the second column, as, for example, there are engines of 100 N.H.P., having a stroke of 48 inches; but since, so far, few have made a practice of thus exceeding the old standard, it is unnecessary to comment upon it further than by saying, that such engines will undoubtedly work well, and with economy both as to coal and working expenses, but will be very costly in construction.

The Cylinder, to calculate the Diameter of.—In this, as in all calculations based on *power*, it is better to deal with I.H.P. than N.H.P.; but, as has already been shown, the latter cannot always be avoided. Hence, when the rule is N.H.P. $= \dfrac{D_h^2 + D_m^2 + D_l^2}{F}$, and R is the ratio of high-pressure to low-pressure cylinder capacity, and R_1 that of medium-pressure to low-pressure.

$$\begin{array}{lccc}
 & \text{Compound.} & \text{Triple.} & \\
\left.\begin{array}{l}\text{Diameter of}\\ \text{H.P. cylinder}\end{array}\right\} = & \sqrt{\dfrac{\text{N.H.P.} \times \text{F}}{1 + \text{R}}} & \sqrt{\dfrac{\text{N.H.P.} \times \text{F}}{1 + \dfrac{\text{R}}{\text{R}_1} + \text{R}}} & \\
\left.\begin{array}{l}\text{Diameter of}\\ \text{L.P. cylinder}\end{array}\right\} = & \sqrt{\dfrac{\text{N.HP.} \times \text{F} \times \text{R}}{1 + \text{R}}} & \sqrt{\dfrac{\text{N.H.P.} \times \text{F} \times \text{R}\,\text{R}_1}{\text{R}\,\text{R}_1 + \text{R} + \text{R}_1}} & \Bigg\} \ (1)
\end{array}$$

Example.—To find the diameter of the cylinders of a compound engine of 200 N.H.P. The allowance being 30 circular inches per N.H.P., and the ratio of cylinders 3·5.

Here R = 3·5, and F = 30.

$$\text{Then diameter of H.P. cylinder} = \sqrt{\frac{200 \times 30}{1 + 3{\cdot}5}} = 36{\cdot}5 \text{ ins.}$$

$$\text{and} \quad ,, \quad \text{L.P.} \quad ,, \quad = \sqrt{\frac{200 \times 30 \times 3{\cdot}5}{1 + 3{\cdot}5}} = 68{\cdot}3 \text{ ins.}$$

If the rule $\text{N.H.P.} = \frac{d_2 + D_2}{F_1} \sqrt[3]{S}$ is to be observed,

$$\left.\begin{array}{l}
\text{then diameter of H.P. cylinder} = \sqrt{\dfrac{\text{N.H.P.} \times \text{F}_1}{(1 + \text{R})\,\text{S}^{\frac{1}{3}}}}\\[2ex]
\text{and} \quad ,, \quad \text{L.P.} \quad ,, \quad = \sqrt{\dfrac{\text{N.H.P} \times \text{F}_1 \times \text{R}}{(1 + \text{R})\,\text{S}^{\frac{1}{3}}}}
\end{array}\right\} \quad - \quad (2)$$

Example.—To find the diameter of the cylinders of an engine of 60 N.H.P., the stroke to be 27 inches, and the allowance 100 per N.H.P.; cylinder ratio, 3·75.

Here $F = 100$; $R = 3{\cdot}75$; $\sqrt[3]{S} = 3$.

$$\text{Then diameter of H.P. cylinder} = \sqrt{\frac{60 \times 100}{(1 + 3{\cdot}75)\,3}} = 20{\cdot}5 \text{ ins.}$$

$$\text{and} \quad ,, \quad \text{L.P.} \quad ,, \quad = \sqrt{\frac{60 \times 100 \times 3{\cdot}75}{(1 + 3{\cdot}75)\,3}} = 39{\cdot}7 \text{ ins.}$$

Example.—If the stroke of the engine in the first example is to be 48 ins. to find the diameter of cylinders.

By reference to Table xiii., the standard stroke for 200 N.H.P. is 39 ins.

$$\text{Hence the diameter of H.P. cylinder} = \sqrt{\frac{39}{48} \times 36{\cdot}5^2} = 33 \text{ ins.}$$

$$\text{and} \quad ,, \quad \text{L.P.} \quad ,, \quad = \sqrt{\frac{39}{48} \times 68{\cdot}3^2} = 61{\cdot}6 \text{ ins.}$$

If the calculations for diameter of cylinders are to be made from the indicated horse-power, and the power is equally divided between the cylinders, whose number is *n*; let R be the number of revolutions per minute; S the stroke *in feet*; p_m the mean pressure in pounds per square inch.

$$\text{Area of piston} \quad - \quad - \quad = \frac{(\text{I.H.P.} \div n) \times 33{,}000}{p_m \times \text{R} \times \text{S} \times 2}.$$

Then diameter of cylinder $= \left.\begin{aligned} &\sqrt{\frac{(\text{I.H.P.} \div n)\,33{,}000}{p_m \times \text{R} \times \text{S} \times \frac{\pi}{4} \times 2}} \\ &= \sqrt{\frac{\text{I.H.P.} \times 21{,}000}{n \times p_m \times \text{S} \times \text{R}}} \end{aligned}\right\}$. (3)

Example.—To find the diameter of the two cylinders of a paddle engine which is required to develop 1200 I.H.P. when working at 30 revolutions per minute; the stroke being 5 feet and the mean pressure 30 lbs.

Diameter of each cylinder $= \sqrt{\frac{1{,}200 \times 21{,}000}{2 \times 30 \times 5 \times 30}} = 53$ ins.

Example.—To find the diameters of the three cylinders of an expansive screw engine to develop 1500 I.H.P., when working at 80 revolutions per minute; the stroke being 3 feet, and the mean pressure 25 lbs.

Diameter of each cylinder $= \sqrt{\frac{1{,}500 \times 21{,}000}{3 \times 25 \times 3 \times 80}} = 41{\cdot}8$ ins.

Example.—To find the diameters of the two cylinders of a compound engine to develop 1000 I.H.P., when working at 70 revolutions per minute, the stroke being 42 inches, and the mean pressures 45 lbs. in the high-pressure cylinder, and 12 in the low-pressure cylinder.

Diameter of H.P. cylinder $= \sqrt{\frac{1{,}000 \times 21{,}000}{2 \times 45 \times 3{\cdot}5 \times 70}} = 30{\cdot}8$ ins

„ of L.P. „ $= \sqrt{\frac{1{,}000 \times 21{,}000}{2 \times 12 \times 3{\cdot}5 \times 70}} = 59{\cdot}8$ ins.

Example.—To find the diameters of the cylinders of a compound engine having one high-pressure and two low-pressure, to develop 6000 I.H.P., when working at 55 revolutions per minute, the stroke being 72 inches, and the mean pressures 35 lbs. in the high-pressure cylinder, and 20 lbs. in each of the two low-pressure cylinders.

Diameter of H.P. cylinder $= \sqrt{\frac{6{,}000 \times 21{,}000}{3 \times 35 \times 6 \times 55}} = 60{\cdot}3$ ins.

„ of each L.P. „ $= \sqrt{\frac{6{,}000 \times 21{,}000}{3 \times 20 \times 6 \times 55}} = 79{\cdot}8$ ins.

Example.—To find the diameters of the cylinders of a triple-compound engine to develop 5000 I.H.P., when working at 80 revolutions per minute, the stroke being 60 inches, and the pressures in high-pressure cylinder 65 lbs. in mean-pressure cylinder 28 lbs., and in the low-pressure cylinder 11 lbs.

Diameter of H.P. cylinder $= \sqrt{\frac{5{,}000 \times 21{,}000}{3 \times 65 \times 5 \times 80}} = 36{\cdot}7$ ins.

„ of M.P. „ $= \sqrt{\frac{5{,}000 \times 21{,}000}{3 \times 28 \times 5 \times 80}} = 55{\cdot}9$ ins.

„ of L.P. „ $= \sqrt{\frac{5{,}000 \times 21{,}000}{3 \times 11 \times 5 \times 80}} = 89{\cdot}2$ ins.

Main Steam Pipe.—The main steam pipe, which supplies a cylinder with steam, should be of such a size that the mean velocity of flow through it does not exceed 8000 feet per minute. When this is not exceeded, the loss of pressure between the boiler and the valve-chest is very slight indeed. If, however, the valve-chest is large, and the cut-off in the cylinder is before half-stroke, the area of transverse section of this pipe may be smaller than given by the above rule, inasmuch as the piston speed is below the mean velocity at the early part of the stroke, and the space in the steam-chest acts as a reservoir for steam, so as to keep up a steady supply during admission. If the space is not less than one-half the volume swept through by the piston at cut-off, the velocity of steam in the pipe may be *assumed* to be 9000 for engines of 150 N.H.P. to 250 N.H.P., and 10,000 for those above that power; for smaller engines, owing to the comparatively larger resistances of small pipes, it is not advisable to take a higher speed than 8000. On the other hand, if the cut-off is later than half-stroke, and the valve-box small, the assumed velocity should be at least 10 per cent. less than that given above.

Taking 8100 feet as the mean velocity, S the mean speed of piston in feet per minute, and D the diameter of the cylinder, then,

$$\text{Diameter of main steam pipe} = \sqrt{\frac{D^2 \times S}{8100}} = \frac{D}{90}\sqrt{S}.$$

Example.—To find the diameter of the main steam pipe to a cylinder 45 inches diameter and 60-inch stroke, the revolutions at full speed to be 60 per minute.

Here $S = 2 \times 5 \times 60 = 600$, and $D = 45$ inches. Therefore,

$$\text{Diameter of main steam pipe} = \frac{45}{90}\sqrt{600} = 12{\cdot}25 \text{ ins.}$$

When the main steam pipe is abnormally long, as is the case with large ships with long boiler rooms, the steam pipe should be somewhat larger.

Area through Stop and Throttle Valves.—Although the loss of pressure at the valve-box is often attributed to want of sectional area in the main steam pipe, it is more frequently due to contracted area past these valves. The friction through a number of small openings is considerably more than through one of an area equal to the collective areas of those openings, especially if the perimeters of the latter largely exceed that of the single opening, and the "loss of head" will be large if due allowance is not made. For this reason there should always be an excess of area around valves and other obstructions to the free passage of steam, and the passages leading to and from them should be as easy as possible, so as to avoid violent changes both of direction and velocity of flow.

Steam Ports and Passages.—Since, in most engines, the steam has to exhaust through the same ports and passages by which it was admitted, their size must be governed by the proper flow of emission, rather than of admission. The area of section of steam ports should be such that the mean velocity of flow should not exceed 6000 feet per minute. The ports may be somewhat larger than the section of the passages when certain kinds of valves are used, which will be dealt with later on; but, as a rule, they have the same area as the sectional area of the passages. To avoid excessive *clearance*, the capacity of the passages should be as small as possible consistent with free flow of steam, and as this depends greatly on their sectional area, the reduction in capacity can only be attained by making them as short as possible. Not only will the evils arising from *clearance* be avoided, but the loss through resistance be very materially lessened by shortening the distance between the valve face and cylinder.

Area of steam ports and of section through passages

$$= \frac{\text{Area of piston} \times \text{speed of piston}}{6000}$$

$$= \frac{(\text{Diameter of cylinder})^2 \times \text{speed of piston}}{7636}.$$

Opening of Port to Steam.—It is advisable so to design the valve, &c., that the opening for admission of steam to the cylinder is sufficient to avoid any serious loss by "wire-drawing;" but in actual practice, unless special gearing is designed so as to give a quick motion to the valve at the instant of cut-off, there is very considerable loss of pressure shown on the indicator-diagram; and, what is worse still, from deficient opening, the loss is generally not limited to the period of cut-off, but during the whole time of admission. The ordinary valve gears do not give that quick motion, either at opening or at cut-off, which is such a desideratum. Separate expansion valves and special valve-gearings admit of such a motion, and consequently the opening to steam with them may be smaller than when cut-off is effected by the ordinary slide-valve and link-motion.

Hence, when only common valves and gear are to be used, the area of opening to steam when at its greatest should be such that the mean velocity of flow does not exceed 10,000 feet per minute. When expansion valves, or special valve-gearing, is used, the mean velocity may be assumed at 12,000 feet, although it is better to give such an amount of opening, when possible, that the velocity shall not exceed 10,000 feet. In actual practice the amount of opening is often much less than that given by the above rules, but it always results in loss of pressure in the cylinder throughout, and excessive "wire-drawing" previous to cut-off.

Exhaust Passages and Pipes.—The area of section of exhaust passages should be such that the mean velocity of steam does not exceed 6000 feet per minute, and if the distance from the cylinder to the condenser is comparatively great, a somewhat larger area is advisable. There should not be a greater difference than 1½ lbs. between the pressure in the cylinder and that in the condenser when exhausting.

The exhaust passages from the high-pressure cylinder of a compound engine to the next cylinder should be such, that the flow of steam does not exceed 5000 feet per minute, in order that the difference between the pressure during admission in the medium-pressure cylinder and exhaust in the high-pressure cylinder may not be excessive, and also from the medium-pressure to the low-pressure cylinder the nominal rate of exhaust should not exceed 5000 feet. The pressure in the receivers is not sensibly constant, as it is in the condenser, being subject to sudden fluctuation when the high-pressure valve opens to exhaust and the medium-pressure valve opens to *lead*, and so on.

The following table gives the relation between the various passages, &c., and the piston in accordance with the foregoing rules, and is based on the assumption of a mean velocity of flow of 8000 feet per minute for the main steam pipe, 12,000 for opening to steam, and 6000 for exhaust; A is the area of piston, and D is its diameter:—

TABLE XIV.

Piston Speed. Feet per Minute.	Diameter Main Steam ÷ D.	Diameter* Exhaust ÷ D.	Area Main Steam ÷ A.	Opening to Steam ÷ A.	Area* Exhaust ÷ A.
200	0·158	0·182	0·025	0·0167	0·0333
250	0·177	0·204	0·0313	0·0208	0·0417
300	0·194	0·223	0·0375	0·0250	0·0500
350	0·209	0·241	0·0437	0·0292	0·0583
400	0·224	0·258	0·0500	0·0333	0·0667
450	0·237	0·274	0·0562	0·0375	0·0750
500	0·250	0·288	0·0625	0·0416	0·0833
550	0·262	0·302	0·0687	0·0458	0·0917
600	0·274	0·316	0·0750	0·0500	0·1000
650	0·285	0·329	0·0812	0·0541	0·1083
700	0·296	0·341	0·0875	0·0583	0·1167
750	0·306	0·353	0·0937	0·0625	0·1250
800	0·316	0·365	0·1000	0·0667	0·1333
850	0·326	0·376	0·1062	0·0708	0·1417
900	0·335	0·387	0·1125	0·0750	0·1500
950	0·344	0·397	0·1187	0·0791	0·1583
1000	0·353	0·400	0·1250	0·0833	0·1667

TABLE XIVa.—WEIGHT OF STEAM IN LBS. AT 100 LBS. PRESSURE DELIVERED THROUGH PIPES WITH A DROP OF 1 LB. ONLY.

By E. C. Sickles.

Diameter of Bore.	Length of Pipe.						
	25 Ft.	50 Ft.	75 Ft.	100 Ft.	125 Ft.	150 Ft.	175 Ft.
1 inch, - -	4·40	3·10	2·54	2·20	1·96	1·79	1·66
1¼ ,, - -	9·76	6·90	5·63	4·88	4·36	3·93	3·69
1½ ,, - -	13·04	9·29	7·69	6·52	5·83	5·33	4·95
2 ,, - -	30·54	21·60	17·60	15·20	13·60	12·50	11·58
2½ ,, - -	50·80	35·90	29·30	25·40	22·70	20·80	19·20
3 ,, - -	...	65·17	...	46·00	...	...	35·00
3½ ,, - -	...	98·30	...	69·50	...	...	52·70
4 ,, - -	...	138·1	...	97·60	...	...	73·80
4½ ,, - -	...	187·9	...	132·9	...	...	102·0
5 ,, - -	...	255·6	...	180·7	...	...	136·8
6 ,, - -	...	419·4	...	296·5	...	...	229·1
7 ,, - -	...	618·8	...	437·0	...	...	330·0
8 ,, - -	...	890·0	...	629·0	...	...	472·0
9 ,, - -	...	1206	...	553·0	...	...	644·0
10 ,, - -	...	1592	...	1126	...	...	851·0
11 ,, - -	...	2046	...	1447	...	...	1093
12 ,, - -	...	2575	...	1887	...	...	1428

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* If the exhaust is to the next cylinder of a compound engine, the area should be 20 per cent. larger.

In naval and other ships the engines of which are only occasionally, and for short periods, run at full speed, so that high efficiency at those times is not of first consideration, the exhaust-pipe to the condenser may be of such a size that the flow is 8000 feet in large engines and 7000 feet in smaller ones; the exhaust-pipes from cylinder to cylinder may be correspondingly reduced, as may also the ports, &c.

Cylinder Liner.—In order that a suitable material may be supplied to resist the rubbing action of the piston without wearing away, and one that shall be capable of taking and retaining a polished surface, so as to minimise the friction of the piston, and which, when worn, may be easily and cheaply renewed, an inner bush or false barrel is fitted, usually called the *cylinder liner*. This liner should be made of a hard, close-grained metal having considerable strength, but not so hard as to resist the action of a cutting tool or file, and capable of taking and keeping a polish when rubbed by the piston-rings lubricated with soft water; it should also be such that the expansion caused by heat is very nearly the same as the cast iron of which the cylinder itself is made. It is usual to make these liners of cast iron, strengthened, closed, and hardened by mixing with it certain kinds of pig iron, or by the addition of a small quantity of steel (*vide* Chap. xxii.). The Admiralty prefer, for horizontal engines, the liners to be made of forged steel, hammered out to a proper size for boring; and some engineers use cast steel. Although the steel gives good results, it can be equalled by the specially-made cast iron, so far as good wearing is concerned, but, of course, it far exceeds cast iron in tensile strength; this latter quality is necessary to a higher degree for the horizontal engine than for the vertical engine, so that the Admiralty are justified in going to the expense of the steel, more especially as it enables them to fit much lighter liners than would be admissible if made of cast iron. In the merchant service, with the vertical engine the cast-iron liner does exceedingly well, and is not likely to be superseded by steel, even if this material can be manufactured much cheaper than at present. Liners are usually made with an inside flange at the bottom end (fig. 29), which fits into a recess in the cylinder end, and is secured there by *screw-bolts*. The upper end is turned for a few inches, so as to fit tightly into the cylinder shell at that part. The joint at the cylinder bottom is made with red-lead paint, while leakage between the liner and the cylinder shell is prevented at the other end by stuffing a few rounds of gasket, rope, or Tuck's packing into a recess formed for that purpose, and preventing it from coming out by securing a flat wrought-iron ring *to the liner* so as to cover the packing. Sometimes in lieu of a stuffing-box, the outer edge of the liner and the edge of the turned part of the cylinder shell are champhered so as to form a groove; into this groove a turn of Tuck's packing or asbestos rope is pressed with a ring as before. Some engineers, preferring to rely on metallic contact, turn a slight recess instead of champhering the edge of the liner, and *caulk into it* a strip of soft copper. The Admiralty method of making a steam-tight joint at the outer end of the liner is by means of a flat copper ring covering the joint and secured to the liner, as also to the cylinder by means of iron rings and screw bolts. The copper ring is deeply grooved between these iron rings so as to permit of a slight movement of the liner endways with respect to the cylinder (*vide* fig. 29*a*). The liners are sometimes secured without a flange at the bottom, by screwing studs through the cylinder shell and liner, and making the ends steam-tight as before.

The space between the liner and shell should not be less than 1 inch, and may be filled with steam so as to heat the steam during expansion. If the cylinder has to be jacketed, this is really a better plan of doing it than by casting the cylinder and inner cylinder together, as was very generally done formerly. Independently of the advantage derived from the harder metal of which the liner may be made, compared with that which is suitable for so intricate a casting as a cylinder, there is another very great advantage to the manufacturer. Since it is a necessity that the walls of the cylinders be sound and free from sponginess, as well as blow-holes, a casting has often to be condemned for a defect which in no way detracts from its strength or usefulness, excepting that it does not admit of the piston working on it steam-tight. If a liner is to be fitted, a little sponginess, or even a blow-hole, is of no consequence, and therefore the extra cost of fitting a liner does not, as a rule, exceed the reasonable premium which would be allowed for assuring good and sound castings; and this is especially so in the case of large cylinders.

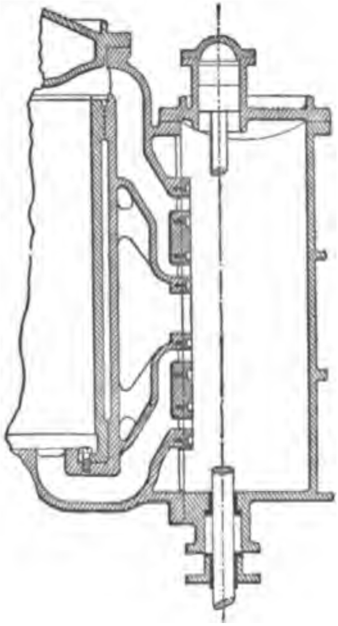
Fig. 29.—Section through Cylinder.

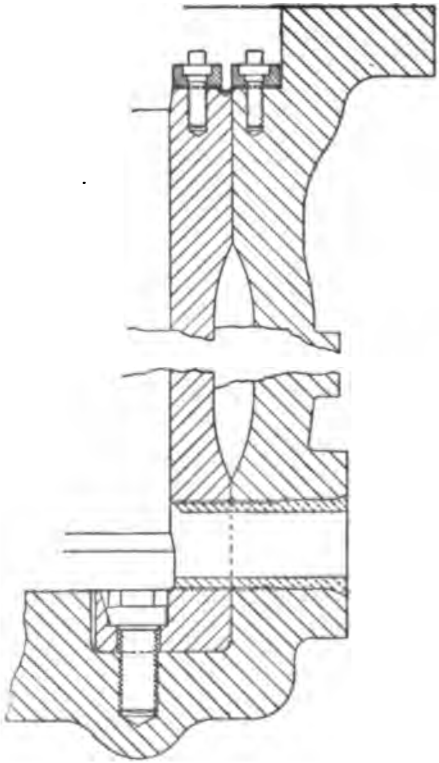
Fig. 29*a*.—Admiralty Method of Fitting Liners.

False Faces.—For the same reason that liners are fitted to the cylinders, the cylinder face needs a false face. This is usually made of hard, close-grained cast-iron, of the same quality as the liner, and secured to the cylinder (fig. 29) by brass screws having *cheese* heads sunk in a recess, so as to be considerably below the surface. Care should be taken to lock these screws, so that they cannot slack back; the simplest way of doing this is to cut a slight nick in the side of the recess, and caulk or drift the metal of the screwhead into it, after the screw is tightened in place.

False faces were sometimes made of hard gun-metal or phosphor-bronze, especially in the engines of warships. The superior strength of these metals over cast iron admits of the face being much thinner, but besides being much more expensive, there is great risk of damage to the cylinder itself, owing to the greater expansion by heat of the bronzes; and they never permit of such a good working surface as does cast iron; and even if this danger is slight, some difficulty has been experienced in keeping the joint between the two metals steam-tight.

By connecting the recesses for the screwheads with grooves cut in the face, the rubbing surfaces are well lubricated, and a considerable amount of relief given to the valve itself by the reduction of the effective pressure on it, caused by the steam flowing through these grooves, &c.

The corners of the ports, both in the false face and cylinder face, should be well rounded, as the casting is very apt to crack if they are sharp.

The Width of the Steam Ports, in the direction parallel to the cylinder bottom, is usually 0·6 to 0·8 of the diameter, but engines of longer stroke than usual require a larger proportion than this to obtain the necessary port area without having excessive length (measured in direction parallel to the axis). It is obvious that, at the cylinder bore, the width of port cannot exceed the diameter, and must really be somewhat less; in actual practice it seldom exceeds 0·8 of the diameter; but at the cylinder face it may, and sometimes does, exceed the diameter, the length being such that the area of section of the passage is uniform throughout.

Piston Valves.—If the very broad cylinder face is bent into the form of a cylinder, there will be the same area of orifice, while the space occupied in direction of the width of the port is less than one-third of that required for the flat face. The valve for such a face must be cylindrical, or composed of two circular discs or pistons having the same depth of edge as there would be of bearing surface at each end of the ordinary slide-valve. Such a valve (fig. 88 in Chap. xiii.) is called a *piston-valve*, and besides possessing the advantage of occupying little space, has the more valuable one of being free from lateral pressure, requiring no balancing or relief, and moving with the least resistance of any slide-valve. For these reasons, the piston-valve is an exceedingly good form when high pressures of steam are used and for very large engines. It is a very general thing among some engineers to fit a piston-valve to the high-pressure cylinder of compound engines of all sizes, and nearly all engineers fit a piston-valve to the high-pressure cylinder of triple-compound engines of all sizes; many makers also fit the medium-pressure cylinder with piston-valves, and a few fit them to all three cylinders, especially of large engines, and of engines running at high revolutions, as in torpedo boat destroyers. The earliest compound engines of Woolf, made in the early years of the nineteenth century, had piston-valves.

Double-Ported Valves.—Although there is of necessity only one opening of the steam passage into the cylinder, there may be two or more openings *through the cylinder face* into the steam passage. The combined area of these openings need not materially exceed that of the section of the passage, and usually only equals it; but as each will be open to steam by the same amount as the single port, if the valve has the same travel, lap, &c., the total opening is in this case double that of the single port for a double-ported face, and treble for a treble-ported face. When the face is treble-ported, the valve is generally arranged so as to admit steam through all three ports, but to exhaust through two only, as there is seldom any difficulty in getting full opening to exhaust.

Steam Jackets.—It is not necessary here to enter into the question of the economy of steam-jacketing. If the economy is doubted, it is, at least, certain that it admits of the cylinders being gently warmed before starting, without moving the working parts. It was customary at one time to form a jacket around the cylinder by *casting it* with two thicknesses of metal; but this was often inconvenient, and always risky to the moulder. Now, when a jacket is desired, a loose liner is fitted. When the engines are of short stroke, the ends present almost as large a surface to the steam as do the walls, and should be jacketed if the jacketing is to be effective. For strength of structure, too, all large cylinders should have hollow bottoms and covers, as stiffening them with webs is not always sufficient, and is in some cases of iron castings even a source of danger. If the pressure is on the same side as the webs, they are safe and do add to the strength of structure; if on the opposite side, then they are in tension, and their outer edge liable to extreme tension; so that if there be a nick or other such defect from which to start a crack, or if subject to a sudden application of the stress, the outer edge is apt to crack, which will develop and spread into the main body of metal, finally causing serious damage. This arises from the fact of cast iron possessing so low a power of resisting stress in tension, compared with its power against compression. Care should be taken to thoroughly drain the steam jackets, and to this end no webs should so be placed as to stop the flow of water to the drain-cocks. The steam supplied to the jackets of the low-pressure cylinder should not much exceed in pressure that in the receiver; for this purpose, a reducing valve is fitted between the boiler supply pipes and the jacket.

Boring Holes.—The diameter of the boring hole depends generally on the size of the boring bar employed, and should not be less than one-fifth the diameter of the cylinder. When there is a single piston-rod, the stuffing-box is formed in the cover of the boring hole. When there are two or more piston-rods, the boring hole should be large enough to admit a man; and, therefore, not less than 14 inches diameter, and, when possible, 16 inches diameter. The doors are sometimes made with the flange fitting into a recess *inside* the cylinder, so that the piston-rod may be drawn from the cylinder with the piston; when this is required, the boring hole must be of sufficient diameter to admit of the piston-rod end drawing through it.

Auxiliary Valves.—To render engines handy—whose main valves cut off at a somewhat early period of the stroke, and compound engines having only one cylinder, into which steam is admitted direct from the boiler—so that they may be started from any position of the cranks, it is necessary to arrange the gear so that the valves may be worked by hand, or else to fit smaller valves, which may be readily worked by the engineer; these are called auxiliary valves, and should have a port area equal to 0·002 the area of the piston. It is only usual to fit such valves to the low-pressure cylinder of compound engines having a cut-off not earlier than half-stroke; when the cut-off is earlier than this, they are fitted to both cylinders, and are usually of the same size—viz., the port area 0·002 the area of low-pressure piston. A careful engineer having an auxiliary valve to the high-pressure cylinder need never use that on the low-pressure cylinder.

Three-crank triple-compound engines are often fitted with an auxiliary slide valve to the low-pressure cylinder, but it is not necessary; pass cocks or valves to admit steam to the medium-pressure and low-pressure valve-boxes are quite sufficient to ensure handiness.

These auxiliary valves are usually only flat plates, without even an exhaust cavity; but for large engines they should be piston valves, or other form of balanced valve.

Escape or Relief Valves.—These are simply spring-loaded safety valves, to allow of the escape of water caused by priming or condensation when the piston presses it to one end of the cylinder. They are fitted to each end of the high-pressure cylinders of all marine engines of 30 N.H.P. and upwards. It is sufficient in vertical engines if there is only an escape valve at the bottom of the medium-pressure and low-pressure cylinders. The diameter of these valves should be one-fifteenth the diameter of the cylinder (the low-pressure of a compound engine being taken). In the Navy it is usual, with horizontal engines, for all large cylinders to have a *pair* of escape valves at each end.

Drain-Cocks.—These should be placed wherever any water is likely to accumulate in the cylinder and casings, and should be 0·4 the diameter of the escape valves. They should be connected to a pipe leading into the condenser bottom; for if led to the bilge the engine-room is filled with steam when open, and the receiver and low-pressure cylinder will seldom drain—in fact, during the greater part of the stroke, instead of letting water out, they let air into the low-pressure cylinder and spoil the vacuum. Care should be taken when the drain-pipe is led to the condenser that the water, &c., does not impinge on the tubes, or even on the condenser sides, so as to do serious damage.

Receiver Space.—The space between the valve of the high-pressure cylinder and that of the medium-pressure cylinder, and that between the valves of the medium-pressure and the low-pressure cylinders, should be from 0·6 to 1·1 times the capacity of the exhausting cylinder, when the cranks are set at an angle of 120°. When the cranks are opposite or nearly so, this space may be very much reduced. The pressure in the medium-pressure receiver should never exceed 0·7 the boiler pressure, and is generally much lower than this. It is usual to fit a safety valve to the low-pressure receiver, loaded by weight or spring to a pressure of 20 to 30 lbs. per square inch; otherwise, owing to the large flat sides between the two cylinders, and in the valve box when a flat valve is employed, great risk of explosion would be run. This safety valve is usually of the same size and design as the cylinder escape valves. The receivers of three-crank compound engines need not be larger than the above, as the cranks are usually at angles of 120°; in the case of triple-compound engines with the medium-pressure leading the high-pressure, a smaller receiver will do.

Column Facings and Feet.—It was very usual at one time to form only facings for the jointing of the cylinder to the frames and columns; but as this necessitated the use of studs, or else driven bolts with the heads inside the cylinders, it is now abandoned, distinct projections or feet being cast to the cylinder bottom, having flanges corresponding to those on the columns or frames, so that they may be connected by driven bolts, which are always accessible. The only objections to this method are, that it is more expensive to mould, and a certain amount of risk is run of getting the casting sound and strong where the feet meet the main casting. The former should be disregarded in considering so important a part as the cylinder, and the latter is always avoided by a good moulder

Great care should be exercised in designing these column feet, for through them the whole force of the steam on the cover (which is equal to that on the piston) is transmitted; and as the load is a recurrent one and always applied suddenly, very ample section of metal should be provided to sustain it. The area of section through these feet should be such that the stress does not exceed 600 lbs. per square inch; that is, the area in square inches is not less than that given by dividing the *maximum* load on the piston in pounds by 600. The webs from the flanges of the feet should be well spread over the cylinder bottom and towards the sides, so as to distribute the strain.

Holding-down Bolts.—The bolts connecting the cylinder to the columns or frames should be such that the stress on them does not exceed 4000 lbs. per square inch, taking the section at the bottom of the thread, and when there is a large number of comparatively small size it should not exceed 3000 lbs. per square inch (*v.* Table xlvi.).

Horizontal Cylinders.—In addition to the facings or feet for connecting to frames, additional feet are necessary for the cylinders of horizontal engines to rest on and be secured to the engine bed. These feet, too, should have webs so arranged as to distribute the strain caused by the reaction from the weight of the cylinder, pistons, &c. The front part of the cylinder should be rigidly bolted down, while the back end, especially of long cylinders, should be *held down* only, and be free to move horizontally when expanded by the heat. But since cast iron will expand only one-tenth of an inch in 8 feet, by an increase of 180° Fahr. of temperature, there is seldom need to make any special provision, beyond boring the holes for the bolts rather larger, or making them slightly oval in the cylinder feet.

Diagonal Cylinders.—The feet of the cylinders of a diagonal engine often serve the double purpose of supporting their weight, and transmitting the strains to the framing; for this purpose they should be as nearly as possible in a plane passing through the axis of the cylinder, and exceptionally long, with good webs, well extended. They should fit into recesses in the framing, and be well keyed against strong fillets at either end, so as to take the shearing strain from the holding-down bolts. If the valve-boxes are on the top sides of the cylinders, either in the diagonal or horizontal engine, care should be taken to form, by means of webs or other device, arches springing from the feet to support their weight, and so avoid distortion of the cylinder bore.

Oscillating Cylinders.—The chief peculiarity of these cylinders is the method of supporting by trunnions, which also serve as steam and exhaust-pipes (*vide* fig. 6). Half the load on the piston is taken on each trunnion and since they are of such ample diameter, it is sufficient to assume that the metal is subject only to shearing stresses, and therefore the area of section should be such that the stress does not exceed 500 lbs. per square inch. The diameter of the trunnions is governed by the size of the exhaust-pipe, since the steam must exhaust through one of them, and it is usual and convenient to make them both of the same size. In the case of a compound oscillating engine, the trunnions of both cylinders should be of the same size, which will depend on the size of exhaust of the low-pressure cylinder. The trunnions of the high-pressure cylinder, being so much larger than is necessary to accommodate the steam-pipe, allows of a space between its outer or working part and the inner part or stuffing-box, which, if left open to the air, is well ventilated, and so prevents the bearing from becoming heated by the steam of high temperature.

The *length* of the trunnion journal or bearing should be such that the pressure per square inch on the area, made by the multiple of its diameter and length, does not exceed 350 lbs.; generally it is from one-third to one-half of the diameter.

The trunnions have interposed between them and the cylinder body a belt, which conveys the steam to and from the valve-boxes. This belt should be very strong and well ribbed to the body of the cylinder, immediately above and below the trunnions, and when the cylinder is fitted with a liner, it is better to form the outer shell in the shape of a beer barrel, so that the belt projects inside and not outside, as it would be were there no liner; the strain from the trunnions is then at once taken by the cylinder sides without the interventions of webs and ribs.

The cylinder facings of oscillating engines should be so set that the edge next the steam entrance should be the nearest point to the cylinder—that is, the plane of the cylinder face touches a cylinder whose axis coincides with the axis of the cylinder-bore at this edge. When this is so, the lead of the steamway into the valve-box is short and easy, and the opening into the exhaust-belt on the side opposite is large, without causing the valve-spindle centre to be unnecessarily far out from the cylinder.

Cylinder Covers.—Like the cylinder end or bottom, the cover has to be strong enough to take the full steam pressure, but as a rule it has no strain to distribute to any other part. The same remarks as to webs, &c., equally apply to the covers, and all above 24 inches diameter of high-pressure cylinders, and 40 inches diameter for low-pressure cylinders, should be made hollow with two thicknesses of metal. Those of vertical engines are better made in that way for all sizes, inasmuch as it is necessary to fill in the spaces between the webs, when they are so made, to prevent the lodgment of water, &c., and it is usual to add a false cover, either polished or cast with a pattern to give a good appearance. This can always be accomplished by casting the covers hollow. The cylinder covers of naval engines are made of cast steel, and formed with ribs having bull-nosed flanges to strengthen them. In small Naval engines, such as those of *destroyers*, the covers, especially of the medium-pressure and low-pressure cylinders, are often made of Stone's or manganese bronze, which has an elastic limit higher than that of ordinary steel, and permits of them being cast much thinner and lighter. Covers when of steel or bronze, are sometimes corrugated to get the necessary stiffness without the complications of ribs. The depth of the cylinder cover at the middle should be about one-quarter of the diameter of the piston for pressures of 80 lbs. and upwards, and that of the low-pressure cylinder cover of a compound engine equal to that of the high-pressure cylinder. Since, however, the size of the piston-rod is the best measure of the pressure on the cover, it is better to so design the cover that its depth at the middle is not less than 1·3 times the diameter of the piston-rod. The depth of the cover at the edge depends on the steam port; a recess being formed for the steam-way, and the inside of the cover otherwise being parallel to the piston.

It is the custom with some engineers to end the cylinder a little beyond the extreme travel of the piston, the steam port-opening being then in the same plane with the cylinder flange; the cover has a large recess in it, and its flange so extended as to enclose the port-opening (fig. 30, p. 149). The advantage of this method is the decreased length and weight of cylinder, and the being able to secure the cover in way of the steam port direct to the main casting, instead of to the comparatively weak bridge of metal across the port. On the other hand, however, the cover occupies considerably more room, and not being of circular form at the flange, cannot be turned there. For larger engines this plan is a very good one, and may be adopted with advantage, but for small ones and those of moderate size it is not so convenient as the older one.

Cylinder Cover Studs and Bolts should be made of the best steel, and of such a size that the stress on them does not exceed 5000 lbs. per square inch of section at the bottom of the thread, as they are subject to severe and sudden shocks when *priming* occurs, and to considerable wear and tear from the frequent removals of the covers for examination of the pistons. In large engines it is usual to fit the cylinder covers with manholes for purposes of examination, as for such large engines larger studs may be fitted, a higher stress is permissible if desired, so that the section may be such that with the maximum pressure there is a stress of 6000 lbs. per square inch. From one or two causes the resistance to pressure on the cover may not be evenly distributed over the whole of the studs, so that a good nominal margin of safety should be allowed in such a very important part; this is especially so when a large number of small studs are fitted; in this case, when of less diameter than 1 inch, the allowance should not exceed 4500 lbs. per square inch. If iron is used, the stress should be 20 per cent. less (*v.* Table xlvi.).

Cylinder Flanges.—The width of the cylinder flange need not exceed three times the diameter of the bolts or studs, but if the former are fitted this allowance is not sufficient to give space for the heads. Studs are now nearly always fitted to marine cylinders in great measure for this reason.

Clearance of Piston.—If both cylinder end, piston, and cover were accurately turned, as is now required by the Admiralty, and the brasses did not wear, a very small amount of space would suffice for clearance between the piston and cylinder end; but as it is usual to leave these parts as they come from the foundry, and the bearings, however well made, do wear in course of time, it is necessary to make due allowance for this. Small engines up to 50 N.H.P. require an allowance of $\frac{1}{8}$ inch at each end for roughness of castings, and $\frac{1}{16}$ inch for each working joint—that is, for any part between the piston and the shaft journals where wear can take place; engines from 50 N.H.P. to 100 N.H.P., $\frac{3}{16}$ inch and $\frac{3}{32}$ inch for each working part; engines from 100 N.H.P. to 200 N.H.P., $\frac{1}{4}$ inch and $\frac{3}{32}$ inch for each working part; engines of 200 N.H.P. and upwards, $\frac{3}{8}$ inch and $\frac{1}{8}$ inch for each working part. Fast running engines should have a larger allowance.

For example, take the case of a vertical direct-acting engine of 120 N.H.P.; the parts which wear so as to bring the piston nearer to the bottom are three—viz., the shaft journals, crank-pin brasses, and piston-rod gudgeon brasses, so that the clearance at *top* will be $\frac{3}{8}$ inch, and the clearance at *bottom* $\frac{3}{8}$ inch + 3 × $\frac{1}{8}$, or $\frac{3}{4}$ inch in all. Here the total clearance in the cylinder is $1\frac{1}{8}$ inch. In a return connecting-rod horizontal engine the same *amount* of clearance was necessary, but the larger allowance at the back or cover end. In a horizontal trunk engine the wear on the shaft journals tends to neutralise that on the connecting-rod ends, so that, if the wear is equal on all of the brasses, an allowance at the front end for one bearing only would be sufficient; but as the shaft journals wear much less than do the others, an allowance for two joints should be made, so that in this case there should be $\frac{3}{8}$ inch at the back or cover end, and $\frac{5}{8}$ inch at the front.

Valve-Box Covers.—The covers are usually formed of a flat plate stiffened by ribs or webs. So long as the stiffening webs are on the same side as the pressure this form is satisfactory, especially when the covers are not very large; but when they are large and it is inconvenient to web them on that side they should be made hollow, with the back rounded like a hog-back girder. These covers in Naval ships are now very generally made of cast steel.

The studs and bolts with which these doors are secured, should be arranged in accordance with the rules laid down for those of cylinder covers.

Small Doors and Covers.—As it is essential to examine from time to time the internal working parts, and as this examination is more for the sake of seeing that everything is in good order, rather than in the expectation of having to execute repairs, and generally there is little time at the disposal of the engineer for these purposes, small doors should be fitted to the large heavy doors, secured with only a few studs, so as to be quickly taken off and refitted. These may for this reason, with advantage, be of cast steel or of steel plate pressed to shape. There should be, when possible, a doorway in the bottom and cover of the low-pressure cylinder, and to the valve casing of the low-pressure cylinder, large enough to admit a man. To the valve boxes of all cylinders there should be peep-holes, through which to ascertain the leads and cut-off of the valves, and to press the valves to the cylinder faces should they have become blown off.

Lagging and Clothing of Cylinders.—All hot surfaces, from which loss of heat may occur by radiation, should be covered with a non-conducting substance. Felt was usually employed and well suited for this purpose, being enclosed in polished teak or mahogany lagging, secured with brass bands wherever in view in the engine-room, and by pine lagging or canvas when not in view. Sheet-iron is frequently used as being more enduring than wood, but unless very carefully fitted and well painted does not look so well as wood, but it lasts much longer.

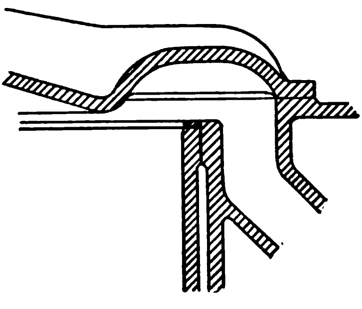

Fig. 30.—Shortened Cylinder with Port in Cover.

Cement and silicate cotton are often specified for by some engineers for the cylinder covering, but they are both very objectionable, on the ground that the dust coming through the lagging from them, owing to the vibration, &c., is very apt to get into the bearings and guides, and cause serious trouble. Asbestos fibre may, however, be used with advantage, especially on the high-pressure cylinder of triple engines, and a cement made of this fibre, &c., has been found very successful; a cement, consisting largely of carbonate of magnesia, has also proved an excellent covering for these and other hot surfaces. It is very necessary to cover up *all* hot surfaces when using steam of such high temperatures as are now common, and considerable gain is obtained by putting muffles on the cylinder covers.

The following rules are for the scantlings of the cylinder and its connections:—

D is the diameter of the cylinder in inches.
p the load on the safety-valves in lbs. per square inch.
p_1 the absolute pressure of steam in the boiler.
f a constant multiplier = thickness of barrel + ·25 inch.

Thickness of metal of cylinder barrel or liner, not to be less than $p \times D \div 3000$ when of cast iron.[*]

$$\text{Thickness of cylinder barrel} = \frac{D}{6000}(p + 50) + 0{\cdot}2.$$

$$\text{„ \quad liner} = 0{\cdot}8 \times f.$$

For purposes of calculation p may be taken as follows:—

H.P. cylinder p = boiler pressure.
M.P. „ triple $p = 0{\cdot}6$ × boiler pressure
M.P. „ quadruple $p = 0{\cdot}7$ × „
M.P.$_2$ „ „ $p = 0{\cdot}45$ × „
L.P. „ „ $p = 0{\cdot}25$ × „
L.P. „ triple $p = 0{\cdot}25$ × „

Thickness of liner when of steel = 0·65 × f.
„ metal of steam ports = 0·6 × f.
„ „ valve-box sides = 0·65 × f.
„ „ „ covers = 0·7 × f.
„ „ cylinder bottom = 1·1 × f, if single thickness.
„ „ „ „ = 0·65 × f, if double „
„ „ „ covers = 1·0 × f, if single „
„ „ „ „ = 0·6 × f, if double „
„ cylinder flange = 1·4 × f.
„ „ cover flange = 1·3 × f.
„ „ valve-box flange = 1·0 × f.
„ „ door flange = 0·9 × f.
„ „ face over ports = 1·2 × f.
„ „ „ „ = 1·0 × f, when there is a false face.
„ „ false face = 0·8 × f, when cast iron.
„ „ „ „ = 0·6 × f, when steel or bronze.

For torpedo boats, gunboats, destroyers, and other such ships where extreme lightness is a necessity, and full power is only developed at intervals and for a short time, the scantling may be reduced by 25 per cent.

Pitch of Studs or bolts in cylinder-cover or valve-box door in inches should not exceed $\sqrt{\dfrac{t \times 100}{p}}$, t being the thickness of the cover or door flange in sixteenths of an inch, p, the pressure per square inch in pounds on it.

Flat Surfaces.—All flat surfaces of cast iron should be stiffened by webs, or stays of some form, whose pitch should not exceed $\sqrt{\dfrac{t^2 \times 50}{p}}$. These webs should be of the same thickness as the flat surface, and their depth at least 2·5 times the thickness.

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[*] When made of exceedingly good material, at least twice melted, the thickness may be 0·8 of that given by the above rules (*v.* Table xlvi.).

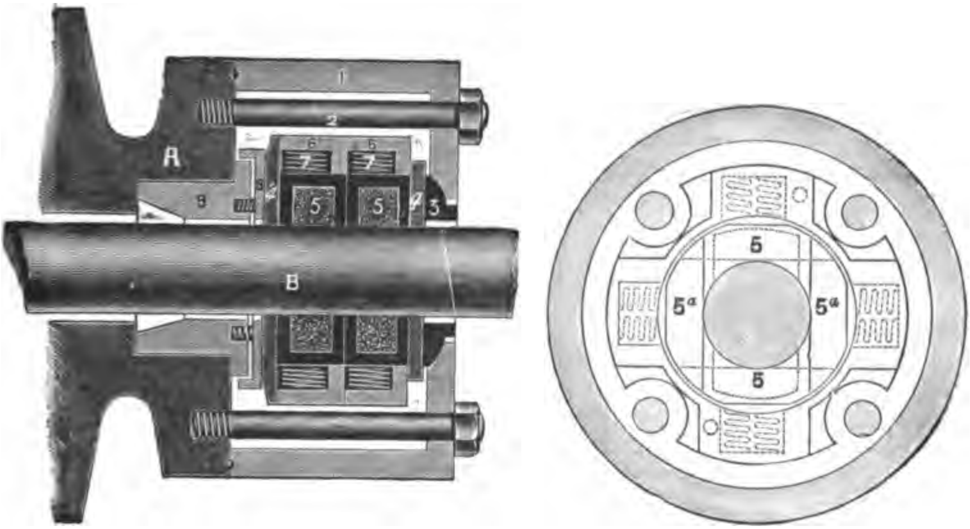


Fig. 31.—The United States Co.'s Packing.

The Cylinder Body or Barrel should be stiffened by external flanges or webs, at about 12 times the thickness of metal apart; these webs should be 1·5 × *f* thick, and stand at least 0·75 × *f* beyond the surface of the cylinder. Some engineers, however, prefer to do without these stiffening webs, and make the cylinder somewhat thicker instead. The low-pressure cylinder, however, differs from the others in size, and in being exposed to external pressure in excess of the internal; therefore it should have webs, especially when of large diameter and without steam jackets.

Stuffing-Boxes and Glands.—For obvious reasons, it is useless giving any definite rules for the sizes of these, as they will differ from different circumstances, and many of the parts do not vary when others are varied. The following Table of sizes gives such as are found in good practice, and will be of more use than any abstract rules:—

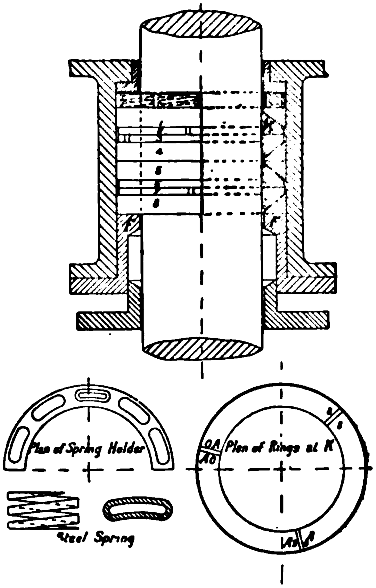


Fig. 31*a*.—Combination Metallic Packing.

STUFFING-BOXES, &c.

TABLE XV.—Stuffing-Boxes for Elastic Packing (Fig. 32).

A	B	C	D_1 50 lbs.	D_2 100 lbs.	D_3 150 lbs.	E	F	
Diameter of Rod.	Width of Packing Space.	Diameter of Box. 1·25 × A + ·55.	Depth of Packing. ·9 × A + ·75.	Depth of Packing. 1·15 × A + ·75.	Depth of Packing. 1·4 × A + ·75.	Overall Depth of Neck-Ring. ·45 A + ·36.	Thickness of Bushes, Gland, and Neck. ·045 × A + ·18.	Number and Diameter of Studs.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
3/4	3/8	1 1/2	1 3/8	1 5/8	1 3/4	3/4	3/16	2 of 3/8
7/8	3/8	1 5/8	1 1/2	1 3/4	2	3/4	3/16	2 ,, 3/8
1	7/16	1 7/8	1 5/8	2	2 1/4	7/8	1/4	2 ,, 1/2
1 1/8	7/16	2	1 3/4	2 1/8	2 3/8	7/8	1/4	2 ,, 1/2
1 1/4	7/16	2 1/8	1 7/8	2 1/4	2 1/2	7/8	1/4	2 ,, 1/2
1 3/8	7/16	2 1/4	2	2 3/8	2 5/8	1	1/4	2 ,, 5/8
1 1/2	1/2	2 1/2	2 1/8	2 1/2	2 7/8	1	1/4	2 ,, 5/8
1 3/4	1/2	2 3/4	2 1/4	2 3/4	3 1/4	1 1/8	1/4	2 ,, 3/4
2	9/16	3 1/8	2 1/2	3 1/8	3 1/2	1 1/4	1/4	2 ,, 3/4
2 1/4	9/16	3 3/8	2 3/4	3 1/2	3 7/8	1 3/8	5/16	2 ,, 7/8
2 1/2	5/8	3 3/4	3	3 3/4	4 1/4	1 1/2	5/16	2 ,, 7/8 3 ,, 3/4
2 3/4	5/8	4	3 1/4	4	4 5/8	1 5/8	5/16	2 ,, 1 3 ,, 3/4
3	11/16	4 3/8	3 1/2	4 1/4	5	1 3/4	5/16	2 ,, 1 3 ,, 7/8
3 1/4	11/16	4 5/8	3 3/4	4 3/4	5 3/8	1 7/8	3/8	2 ,, 1 1/8 3 ,, 7/8
3 1/2	3/4	5	4	5	5 5/8	2	3/8	2 ,, 1 1/8 3 ,, 1
3 3/4	3/4	5 1/4	4 1/4	5 1/4	6	2 1/8	3/8	2 ,, 1 1/4 3 ,, 1
4	13/16	5 5/8	4 1/2	5 1/2	6 3/8	2 1/8	3/8	2 ,, 1 1/4 3 ,, 1
4 1/2	7/8	6 1/4	4 3/4	6	7	2 3/8	7/16	2 ,, 1 3/8 3 ,, 1 1/8
5	15/16	6 7/8	5	6 1/2	7 3/4	2 5/8	7/16	2 ,, 1 3/8 3 ,, 1 1/8
5 1/2	1	7 1/2	5 5/8	7	8 1/2	2 7/8	7/16	4 ,, 1 1/8
6	1 1/8	8 1/4	6 1/4	7 5/8	9 1/4	3	7/16	4 ,, 1 1/8
6 1/2	1 1/8	8 3/4	6 5/8	8 1/8	9 7/8	3 1/4	1/2	4 ,, 1 1/8
7	1 1/4	9 1/2	7	8 3/4	10 1/2	3 1/2	1/2	4 ,, 1 1/4
7 1/2	1 1/4	10	7 1/2	9 3/8	11 1/4	3 3/4	1/2	4 ,, 1 1/4
8	1 3/8	10 3/4	8	10	12	4	9/16	4 ,, 1 1/4
8 1/2	1 3/8	11 1/4	8 1/2	10 1/2	12 3/4	4 1/4	9/16	4 ,, 1 3/8
9	1 1/2	12	8 7/8	11	13 3/8	4 3/8	9/16	4 ,, 1 3/8
9 1/2	1 1/2	12 1/2	9 3/8	11 5/8	14 1/8	4 5/8	5/8	4 ,, 1 3/8
10	1 1/2	13	9 3/4	12 1/4	14 3/4	4 7/8	5/8	4 ,, 1 3/8

In the case of the cylinder, it is usual to make the stuffing-boxes, for uniformity's sake, for the low-pressure and medium-pressure piston-rods of the same depth as that of the high-pressure rod. The stuffing-boxes of the valve-spindles, too, are usually exceptionally deep, on account of their liability to leak, and the trouble of packing them. The packing of the stuffing-boxes in the cylinder-covers of vertical engines is very liable to give trouble with steam of high-pressure from the want of moisture; the lubricant affects only the top layers of packing, and keeps them soft, while the bottom ones get hard and charred. Metallic packings (*v.* figs. 31 and 31*a*) are the best for use with steam of high-pressure, and although some patent vegetable packings work very well, no doubt these latter will all in time be superseded by the former. The metallic packings are generally arranged in a series of hoops of triangular section, the pressure on the rod being caused either by a second set of hoops outside the first, causing a wedging action on the gland being pressed home, or else by an arrangement of springs or spring clips. The newer and better forms are now giving great satisfaction and taking the place of vegetable and asbestos in the high-pressure cylinder, and used by most engineers for the medium-pressure cylinder also. Many engineers fit metallic packing in the low-pressure cylinder glands others strongly object to do so, much preferring good vegetable packing on account of the excessive moisture in this cylinder.

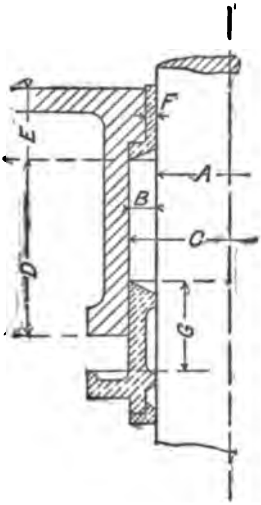


Fig. 32.

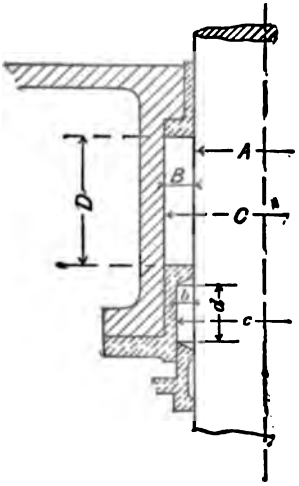


Fig. 32*a*.

CHAPTER VIII.

THE PISTON—PISTON-ROD—CONNECTING-ROD.

The Piston is essentially only a disc, strong enough structurally to withstand the pressure of the steam on it, and fitting steam-tight in the cylinder. The piston in this simple form is seen in the Richard's Indicator, and is often so fitted to small engines.

In the early days of steam-engine construction, when there existed no machine capable of boring out a cylinder, the bore was not perfectly true, nor the sides very smooth, and, consequently, unless some form of elastic packing was interposed between the piston and the cylinder sides, it could not work steam-tight. It was customary to form the piston with a recess on the rim, into which rope or *junk* was coiled, just as is now the custom to do with air-pumps. This packing could not be examined or renewed without drawing the piston from the cylinder, a tedious operation at all times; to remedy this the recess was made without a flange at the top or side of the piston next the cylinder cover, and a false flange or loose ring was bolted to the piston so as to retain the junk packing in place, and admit of its being removed or added to without removing the piston from the cylinder. This ring was called the "junk-ring," and retains that name although junk is no longer used to pack pistons. After a few weeks' work the cylinder was rubbed smooth and fairly true, when the piston would work steam-tight with very little friction, and with steam of low-pressure and temperature the packing lasted a considerable time.

A solid piston, that is, one without packing, is really the best for good working, so long as it remains steam-tight; but as there is always some slight amount of wear, especially when the cylinder is fresh from the boring mill, and leakage past the piston is most serious, more particularly when the engine is standing still, it is necessary to have some means of adjustment, whereby the piston is maintained a steam-tight fit in the cylinder.

In lieu of the vegetable packing, which is not admissible with steam of high pressure, engineers now fit metallic rings, called "packing-rings," in various forms, which are pressed outwards against the side of the cylinder by springs. These rings are maintained in position steam-tight by the junk-ring as of old.

When these metallic rings are once in place so as to fit closely to the cylinder sides, there is no need of further lateral pressure until by wear the piston becomes slack, and steam permitted to pass it. However, nearly all existing pistons are automatic in this respect, and the consequence is that the packing-rings press so tightly on the cylinder sides, that the loss by friction seriously impairs the efficiency of the engine; and it is only when the ring or cylinder is considerably rubbed away, that the piston works with ease. From these causes many really good pistons have been condemned after *having been made* to cause serious damage.

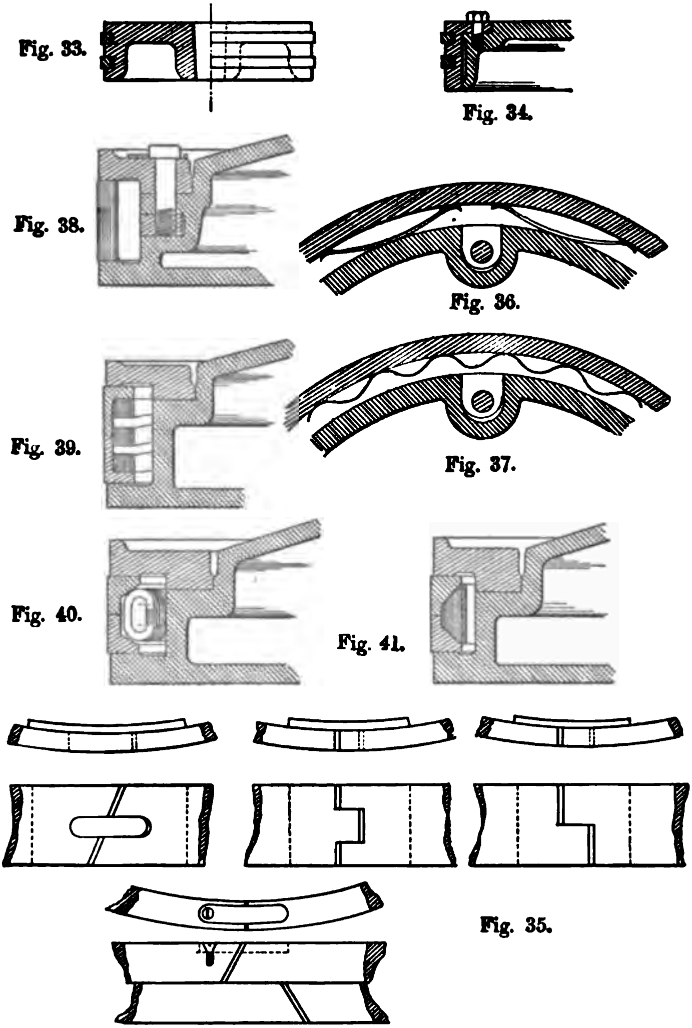
Perhaps the first remove from the primitive piston is to be found in the form usually fitted in locomotives, and generally known as *Ramsbottom's*.

Ramsbottom's Rings (fig. 33).—The late Mr. Ramsbottom, of the L. & N.W.R. Co., was the first to pack pistons by one or more narrow metal rings, turned somewhat larger in external diameter than that of the cylinder bore, and which, after being cut across so as to be capable of being compressed to suit the bore of the cylinder, are fitted into recesses turned in the piston edge. The rings fit easily into these recesses, and as they are so placed that no two of the joints are in a line, the piston is practically steam-tight, and works very well in locomotives and other quick working engines of small size; but for large engines, and engines undergoing the same vicissitudes as those on shipboard, there is an objection to this form of piston. It will be seen that the rings cannot be removed without drawing the piston, and that there is no means of preventing steam from passing where the spring is cut across, besides which the rubbing surface is very small, and the spring is always exerting its maximum effort. The first of these objections is overcome (fig. 34) by fitting a junk-ring, having cast with it a spigot or ring, which goes down into the recess around the piston for the packing-ring and made steam-tight; into grooves turned in the outer surface of this spigot the Ramsbottom rings are fitted.

For small engines these rings are made of steel; for such engines as may be standing unused for many days, some engineers prefer to fit hard brass rings. When for larger engines where the section may be three-quarters of an inch square and upwards, the rings are better of tough and hard cast iron, which works very well indeed.

Common Piston-Rings (figs. 35 to 38) consist only of a single hoop made of very tough, close-grained, cast iron, made on the same principle as the Ramsbottom rings, but fitted between the piston flange and the junk-ring, so as to be free to move laterally steam-tight. This packing ring is usually turned to a diameter about 1 per cent. in excess of that of the cylinder, and either cut across diagonally, or formed so that one end has a tongue fitting into a recess in the other (fig. 35), a brass cover-piece being fitted behind the gap, so as to prevent steam leaking into the space behind the rings. The ring is then fitted to the piston flange steam-tight by scraping both surfaces; the ring is raised by interposing very thin pieces of paper between it and the flange, and the junk-ring is then fitted steam-tight to the piston and packing-ring by scraping, &c. Some makers of pistons profess to turn the piston and rings so accurately as to require no scraping, but it is doubtful if there is economy in the practice if carried to the perfection professed; other engineers prefer to grind the rings tight after coming from the lathe. In whatever way the object is attained is of small moment compared with the necessity of having the ring *perfectly* steam-tight between the flange and junk-ring.

Piston Springs.—When the piston is of comparatively small diameter, the elasticity of the packing-ring itself is sufficient to keep it steam-tight against the cylinder sides for a very considerable time after it is fitted; and even larger rings may be made of sufficient strength to do this, but they would then be open to the same objection as raised against the Ramsbottom rings. The old method of pressing the ring out by means of dished springs or coach-springs, as shown in fig. 36, is now seldom used in new engines; the objections to it are the uneven and unknown pressure exerted, and the reaction of the piston itself, from the fact of the springs pressing on it. It was a very difficult thing to so set every spring that the pressure on the ring was uniform; and the range of action of this form of spring is very limited, so that although the ring might be very tight when first fitted, after a few days running it would be passing steam. The surface exposed to pressure too was small, and the springs were apt to bed themselves into the ring, and in doing so wear through their curved ends. These defects were partially remedied by adding to each one or more subsidiary springs on the principle of coach-springs, but that only tended to aggravate the other evil spoken of —viz., the reaction of the piston itself.



Figs. 33 to 41.—Various Forms of Piston Packings.

When a piston is moving through its course, and guided therein by the rod at one end and the tail rod (or back guides in case of a horizontal engine) at the other, it should be quite free laterally from the packing-ring, which may follow its course freely. When the bore of the cylinder is quite true, and its axis coincides with the line of motion of the piston centre, it is of no consequence if the springs do bear on the piston; but if the cylinder wears somewhat out of truth in either direction, it is important that the spring-ring shall follow the sides of the cylinder freely; it cannot do this if the springs react from the piston body.

Cameron's Patent.—Fig. 37 shows a piston-ring pressed out with a corrugated ribbon of steel; the lateral pressure here is obtained by the resistance of the spring to being bent into a circle, and by the pressure exerted by the corrugations when the ends of the spring are pressed apart. This spring exerts an almost uniform lateral pressure on the packing-ring without touching the body of the piston, and by making the packing-ring comparatively thin, it will adapt itself to the shape of the cylinder when worn. The pressure on the ring can also be easily and nicely adjusted by packing pieces between the ends of the spring. One great advantage of this spring is that it can be fitted to any piston without condemning any of the parts beyond the springs.

Mather and Platt's Patent.—It was found that metallic packing-rings not only wore sideways, but also on the edges, so as to become slack between the flange and junk-ring; a very slight amount of play between these soon causes a very large degree of slackness from the continual concussion on change of motion at every stroke. To obviate this the ring was formed with inside flanges, as shown in fig. 39, and split into two, a spiral hoop, having three or four turns, being coiled inside the rings, whose action is to press the packing-rings outward against the cylinder sides, and up and down against the flange and junk-ring. This form has been generally very successful, and pistons so fitted have worked very well indeed; but there is the objection that no adjustment of the spring is possible, and it is always exerting its maximum effort. The chief part of the elasticity of this spring, however, is exerted in pressing the rings against the flange and junk-ring, and the friction so caused helps to prevent undue pressure on the cylinder side, so that in practice it is not found that there is excessive side pressure when first fitted, nor lack of it when the cylinder is worn. These springs are made of steel, or very strong cast iron, cut out of a ring of either metal. They are also sometimes cast to the form required.

Buckley's Patent consists of two rings, of section as shown in fig. 40; a spiral coil of steel wire is bent into a circle, and inserted between the two packing-rings. Pressure is exerted in the same way as in Cameron's spring, and tends to press the packing-rings both outwards and against the flange and junk-ring. This form of piston is very generally used at the present time, and when properly adjusted works very well. The spring, however, is a very stiff one, and requires but little end pressure to exert a very considerable pressure on the sides of the cylinder.

Prior's Patent.—The rings in this plan are precisely like those of Buckley's piston, but the spring is made of bar steel bent zigzag, so as to form a belt and exert pressure both outwards and sideways in the same way that Buckley's does.

Qualter and Hall's Patent.—This piston has two packing-rings of triangular section, with a third ring inside and between them, as shown in fig. 41, so that on this inner ring being pressed outwards, it exerts a wedge action on the two packing-rings so as to force them against the flange and junk-ring. Coach-springs are employed to press the ring outwards, which are each held in a brass frame having a tapered piece on the back, which fits into a recess in the piston and against a tapered cotter; this cotter can be pressed down by a set screw in the junk-ring, and any required pressure is imparted to the ring through the spring in this way. This piston has therefore the advantage of being capable of adjustment without removing the junk-ring, and the adjustment can be made to a nicety with very little trouble, and also, in case of the engines not being required for some weeks, the pressure on the springs may be relieved until required again. This piston has given very great satisfaction.

Rowan's Patent and MacLaine's Patent each consists of two strong rings of square section, or of U section, pressed outwards by springs at the cross cut, and against the flange and junk-ring by wave springs fitted in a groove between them. This arrangement has been found to work well.

Restrained Packings.—In the early days of piston-making designers were chiefly concerned in providing means for keeping the packing-rings on the cylinder walls; as pressures of steam increased they still experienced the

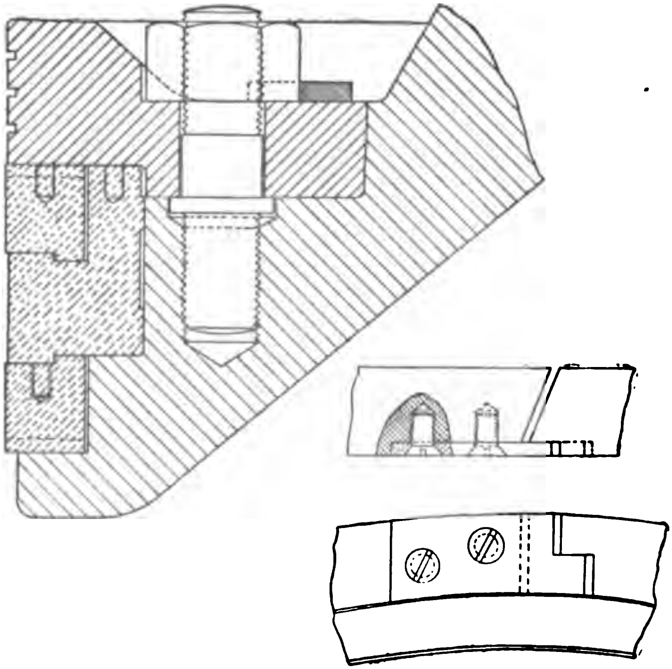


Fig. 42.—Restrained Packing-Rings (Admiralty Plan).

same anxiety lest there should be loss by leakage. To-day we have less anxiety on that score, but a more serious one respecting the loss due to the rings being pressed unduly hard on the cylinder by the steam now used. We cannot keep the steam from getting behind the rings, so it is necessary to restrain them; this was first done by securing the ends or locking them together by the same device that set them. With the very high pressures

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now employed even that is not sufficient, and it is the practice in the Navy to fit solid restraining-rings, as shown in fig. 42, to keep the cut rings in place when fitted, as in fig. 44 (right-hand illustration).

Body of the Piston.—Pistons of small size are usually made of a single thickness of metal without, of course, any stiffening webs or ribs. Pistons of very considerable size have been made of cast steel in this way; they are in the form of a cone, to suit the cylinder end, and by that means have the requisite degree of stiffness; they were originally made in this form to save weight, being for fast-running horizontal engines, but are now used extensively in all fast-running engines. Cast-steel pistons are now frequently used in the mercantile marine in engines of all sizes; chiefly, however, when in those running at high speeds if small and in those of large size. They may be used with advantage in most engines, and are not much more costly than hollow cast-iron ones when of large size. The pistons for torpedo-boats and other very light, fast running machinery are usually made of forged steel, and are very thin at the flange and near it.

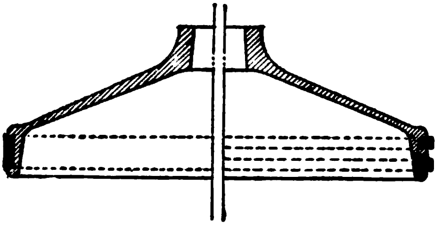

Fig. 43.—Forged-Steel Pistons.

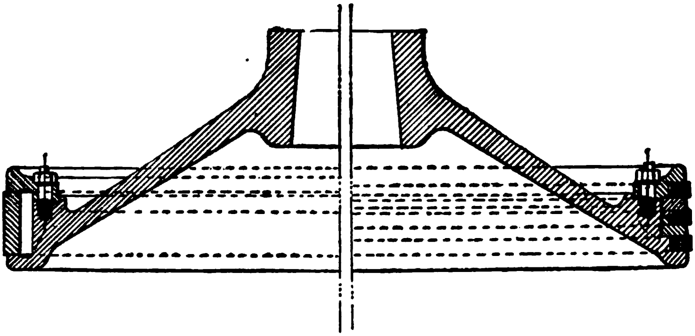

Fig. 44.—Cast-Steel Pistons.

Pistons of ordinary marine engines above 12 inches diameter for a high-pressure, and 24 inches diameter for a low-pressure cylinder, are usually made cellular—that is, with two thicknesses of metal stiffened or connected by ribs and webs, and either by the thickness of metal or by the depth of body made strong enough structurally to safely withstand, not only the load due to steam pressure exerted on it and transmitted to the rod, but also the shocks to which it is liable when priming occurs.

The piston body must be so designed, too, that it may be safely cast, for in the early days of large pistons it was not at all an uncommon thing for a piston to break in cooling, or mysteriously afterwards. For this reason any rules must of necessity be empirical which set out the thickness of metal of the different parts of the body; but care must always be exercised that no one part is too small for the stresses to which it is subject. For example, there must be sufficient metal in the immediate neighbourhood of the piston-rod boss to resist the tendency to force out this part by shearing the metal. Again, the piston may be taken as consisting of a number of sectors, and by considering one of such small sectors loaded with the pressure on its area at the centre of gravity of its figure, the bending moment at any section may be found, and the thickness of metal tried whether it be sufficient for the purpose.

For the section of an ordinary piston having a single rod, the following table gives the multipliers for obtaining the thickness of metal and sizes of the different parts.

Details of Construction of the Ordinary Piston.—Let D be the diameter of the piston in inches, p the maximum effective pressure per square inch on it, x a constant multiplier, found as follows:—

$$x = \frac{D}{50} \times \sqrt{p} + 1.$$

For high-pressure cylinders p may be taken at half the boiler pressure; for medium-pressure cylinders a quarter the boiler pressure; and for low-pressure cylinders half the boiler pressure, divided by ratio of low-pressure to high-pressure cylinders.

HOLLOW CAST-IRON PISTONS.

The thickness of front of piston near the boss - = $0.2 \times x$.
,, ,, ,, ,, rim - = $0.17 \times x$.
,, back ,, - - - = $0.18 \times x$.
,, boss around the rod - - = $0.3 \times x$.
,, flange inside packing-ring - = $0.23 \times x$
,, ,, at edge - - - = $0.25 \times x$.
,, packing-ring - - - - = $0.15 \times x$.
,, junk-ring at edge - - - = $0.23 \times x$.
,, ,, inside packing-ring - = $0.21 \times x$.
,, ,, at bolt holes - - = $0.35 \times x$.
,, metal around piston edge - = $0.25 \times x$.
The breadth of packing-ring - - - - = $0.63 \times x$
,, depth of piston at centre - - - - = $1.4 \times x$.
,, lap of junk-ring on the piston - - - = $0.45 \times x$.
,, space between piston body and packing-ring = $0.3 \times x$.
,, diameter of junk-ring bolts - - - = $0.1 \times x + 0.25$ in.
,, pitch ,, ,, ,, - - - = 10 diameters
,, number of webs in the piston - - - = $\frac{D + 20}{12}$.
,, thickness ,, ,, - - - = $0.18 \times x$.

SOLID PISTONS.

	Cast Steel.	Cast Iron.
Thickness near boss - - - -	= $0.26 \times x$	= $0.52 \times x$.
,, ,, rim - - - -	= $0.15 \times x$	= $0.22 \times x$.

Forged Steel Pistons.

Thickness near boss - - - - - $= 0{\cdot}2 \times x$.
" " rim - - - - - $= 0{\cdot}1 \times x$.

When made of exceptionally good metal, at least twice melted, the thicknesses of cast-iron pistons may be as much as 20 per cent. less than given by the rules; but, on the other hand, if made of other than really good metal, they should be thicker. The piston should be made of good metal always, and for fast-running engines it is better made of steel. The packing-ring is sometimes made much thicker in the part opposite the cut than given above, in order to have sufficient elasticity of itself to press steam-tight against the cylinder; but it is better to let the springs perform their function wholly, and leave the ring to act only as the packing.

Junk-ring Bolts.—When screw-bolts are used to hold the junk-ring in place, they are either screwed into a brass nut let into a recess in the side of the piston (*v.* fig. 36), or else screwed into a brass plug, which has been screwed tightly into the piston. The former plan is most general, and has the advantage that if the bolt thread is torn away the nut can be easily replaced, and owing to the length of body the bolts cannot slack themselves back. Many engineers, however, prefer to screw the bolts directly into the cast iron, making the tapped hole as deep as possible; and although it may be supposed the bolts would set fast by rust, practice has shown that such does not take place, nor do the cast-iron threads wear quickly away.

Studs are often used instead of screw-bolts (*v.* fig. 42), but although, to some extent, possessing advantages over the latter, they are not so convenient, and have all to be withdrawn when any refitting of the packing and junk-rings is necessary.

Safety-rings and Lock Bolts.—The vibration of the junk-ring has a tendency to slack back the bolts, and although it is a rare occurrence to find such a thing happen in a vertical engine, very serious accidents have frequently been caused by the junk-ring bolts getting loose in a horizontal cylinder. To prevent such a casualty the piston bolts of all engines should have a light wrought-iron ring secured to the junk-ring by studs, having square bodies and nuts secured with split-pins; this ring (*v.* fig. 42) fits close to the heads of the bolts, and prevents them then from turning. When studs are used their bodies should be square or with a projecting side, the holes in the junk-ring corresponding in size and form to them, so that, when on, it prevents them from unscrewing; the nuts may be prevented from slackening by a ring, or each stud may have a split-pin through its end.

There are some other methods, but none of them are either so efficient or so inexpensive as the above.

Solid Packings.—In order that the weight of the piston of a horizontal engine may be taken by the broad packing-ring, instead of by the comparatively narrow flange and junk-ring, it is customary and advisable to fit a cast-iron packing between the body of the piston and the packing-ring for about one-third of the circumference in lieu of springs. The pistons of diagonal and oscillating cylinders are also better if fitted in this way.

Piston-rod.—It is usual to have only one rod to each piston of a direct-acting engine, but some manufacturers, to suit a particular style of crosshead and connecting-rod, fit two. The single rod is preferable from practical considerations, even for large engines, because it requires very considerable care on the part of the workman to bore the two holes in the piston, cylinder bottom, and crosshead so *exactly* that the rods will fit into their place without adjustment; the friction of the two stuffing-boxes will be very considerably more than that of the one larger one; the cost of labour will also be nearly double that for the single rod, and there are two stuffing-boxes, which require packing, and two glands demanding attention, instead of one.

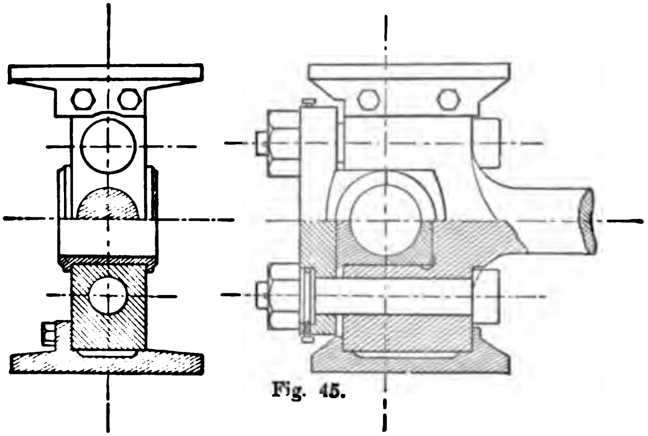

Fig. 45.

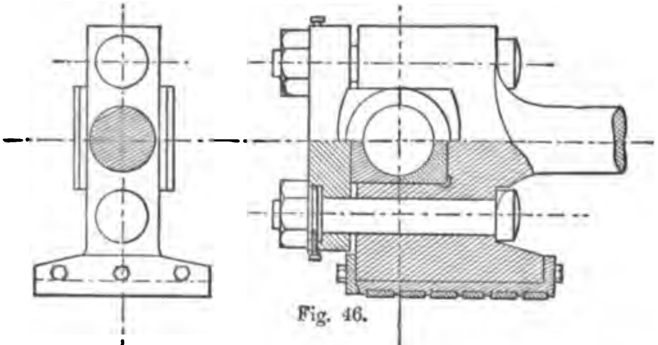

Fig. 46.

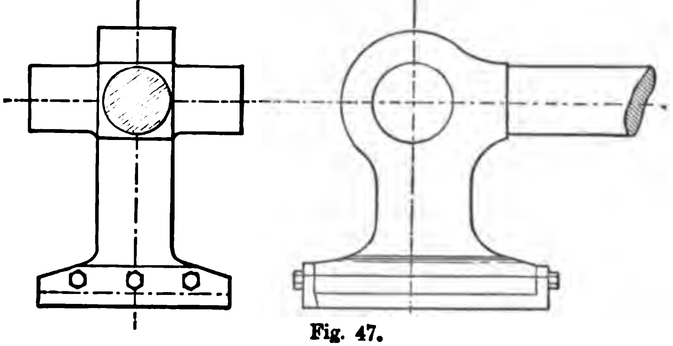

Fig. 47.

Piston-Rod Ends and Guide Blocks, &c.

Return connecting-rod and large steeple engines of necessity require two rods to each piston, and Messrs. Humphrys fitted four rods in the case of the very large pistons of H.M.S. "Monarch," the better to distribute the strain over the piston face, and to admit of a better form of crosshead.

Diameter of Piston-rod.—Since the piston-rod is secured in the piston, and usually well guided at the other end, so that it could not bend without meeting with considerable resistance, it may be treated as a *strut* or column, secured at both ends; but when the outer end fits into a crosshead, which would offer little or no resistance to bending, then the rod must be treated as a column loose at one end and secured at the other.

From Mr. Hodgkinson's experiments and Mr. L. Gordon's investigations, the following are the formulæ for computing the strength of columns:—

(1) For a column fixed at both ends, $P = \dfrac{f \cdot S}{1 + a\dfrac{l^2}{d^2}}$.

(2) For a column loose at both ends, $P = \dfrac{f \cdot S}{1 + 4a\dfrac{l^2}{d^2}}$.

(3) For a column fixed at one end only, $P = \dfrac{2f \cdot S}{2 + 5a\dfrac{l^2}{d^2}}$.

P is the load, l the length of the column in inches, d the diameter in inches, a for solid wrought iron and mild steel $\frac{1}{3000}$, and f 36,000 lbs. per square inch, S being the area of section of the rod in square inches. Taking this value of f in the above formulæ, P is the breaking load; since it is usual to have a factor of safety for all important parts of a marine engine, of at least 6, the value of f should not exceed 6000 lbs., and may be taken generally at 5000 lbs.; but as the piston-rod is liable to great shock, is always working with alternating stresses, and always receives its load suddenly, 3000 lbs. should be taken as the value of f to calculate the diameter. These formulæ are too complicated for general use, but the size of a piston-rod may be *checked* by them easily after having been calculated by an empirical formula.

Since, however, an approximate value may be safely taken for the relation between l and d of ordinary marine engines, the formulæ may be reduced to a very simple form. Hence, since

$$P = \frac{\pi D^2}{4} \times p: \quad \text{and} f S = \frac{\pi d^2}{4} \times f.$$

D being the diameter of the piston, and p the effective pressure on it in pounds per square inch.

$$p \cdot D^2 = f d^2 \div 1 + a\frac{l^2}{d^2}.$$

Let

$$\frac{l}{d} \text{ be represented by } r,$$

Then

$$d = D\sqrt{\frac{p}{f}(1 + a r^2)}$$

taking $f = 3025$.

$$\text{Diameter of piston-rod} = \frac{D}{55}\sqrt{p(1 + a r^2)}. \quad - \quad - \quad - \quad - \quad (1)$$

The following are the values of r :—

Short stroke direct-acting engines, initial pressure	30 lbs.,	$r =$ 7 to 8
Long „ „ „	30 „	$r =$ 8 to 10
Short „ „ „	80 „	$r =$ 6 to 7
Long „ „ „	80 „	$r =$ 7 to 8
Very long „ „ „	80 „	$r =$ 10 to 12
Oscillating engines of short stroke „	30 „	$r =$ 10 to 15
„ „ long „ „	30 „	$r =$ 15 to 18
„ „ medium „ and compound	70 „	$r =$ 10 to 12
Return connecting-rod engines, boiler pressure	30 „	$r =$ 22 to 28
„ „ „	70 „	$r =$ 18 to 23

The following rules, however, will give results sufficiently accurate for all practical purposes.

$$\text{Diameter of piston-rod} = \frac{\text{diameter of cylinder}}{F}\sqrt{p}.$$

The following are the values of F :—

Naval engines, direct-acting, - - -	F = 50
„ return connecting-rod, 2 rods, -	F = 70
Mercantile ordinary stroke, direct-acting, -	F = 45
„ long „ „ - -	F = 42
„ medium „ oscillating, - -	F = 40

NOTE.—*Long and very long*, as compared with the stroke usual for the power of engine or size of cylinder.

D is the diameter of the low-pressure and d that of the high-pressure cylinder of a compound system; d_1, d_2, &c., the diameters of the intermediate cylinders; R is the ratio of low-pressure to high-pressure cylinder volumes; r_1 the ratio of volumes of the first intermediate and high-pressure cylinders; r_2 that of the second intermediate and high-pressure cylinders; if the stroke be the same in each cylinder

$$R = \frac{D^2}{d^2}\,;\; r_1 = \frac{d_1^2}{d^2}\,;\; r_2 = \frac{d_2}{d^2}.$$

The maximum pressures repeatedly applied on the pistons when working full speed are approximately as given by the following formulæ :—Where p_a is the absolute pressure at or near the engines, and may be taken as the load on safety-valve *plus* 15 lbs. for purposes of calculation of sizes of piston-rods, connecting-rods, columns, &c., and their bolts and fittings.

Values of p or maximum effective working pressure in

High-pressure cylinder of a compound engine, - -	$p_a \times 0{\cdot}75$
Low-pressure „ „ „ - -	$p_a \div R \times 1{\cdot}2$
High-pressure „ triple-compound engine, -	$p_a \times 0{\cdot}63$
Medium-pressure „ „ „ -	$p_a \div r_1 \times 1{\cdot}5$
Low-pressure „ „ „ -	$p_a \div R \times 1{\cdot}5$
High-pressure „ quadruple-compound engine,	$p_a \times 0{\cdot}55$
1st medium-pressure cylinder of a quadruple-compound engine,	$p_a \div r_1 \times 1{\cdot}65$
2nd „ „ „ „ „	$p_a \div r_2 \times 1{\cdot}55$
Low-pressure „ „ „ „	$p_a \div R \times 1{\cdot}65$

Except in the case of the triple, with two low-pressure cylinders, whose rods may be and usually are smaller than those of the high-pressure and medium-pressure cylinders. It is at all times desirable that the piston-rod shall move through the stuffing-box without vibration, but especially is this so when metallic packing is used. It is, therefore, a good thing to have the piston-rods larger, rather than smaller, than given by the above rules.

Piston-rod Ends.—It is absolutely necessary that the rod should fit *perfectly* steam-tight into the piston, and also be so tight as not to "draw" in the least when subject to shock. If the rod end were made cylindrical or "parallel," as it is technically called, and fitted in to satisfy the above conditions, it would be very tedious and difficult to get it out again. For this reason principally it is usual to turn the part fitting into the piston "taper" or conical. If the taper is very slight the rod can be easily made a tight fit, but unless formed with a shoulder at the end of the taper, it would in time become so tightly held by the piston as to withstand all attempts at withdrawal, and also there would be at all times a great danger of splitting the piston by the wedging action. If the taper extends the whole depth of the piston, it should be at the rate of $\frac{3}{4}$ inch to the foot; that is, the diameter of the rod at back is less than that at the front by one-sixteenth of the length of taper. Even with so liberal an allowance as this, there is great difficulty often experienced in withdrawing the rod after a few months' work; for this reason some engineers do not extend the taper the full depth of the piston. The most convenient, and at the same time reliable, practice is to turn the piston-rod end with a shoulder of $\frac{1}{16}$ inch for small engines, and $\frac{1}{8}$ inch for large ones, make the taper 3 inches to the foot (fig. 48) until the section of the rod is three-fourths of that of the body, then turn the remaining part parallel; the rod should then fit into the piston so as to leave $\frac{1}{8}$ inch between it and the shoulder for large pistons, and $\frac{1}{16}$ inch when small. The shoulder prevents the rod from splitting the piston, and allows of the rod being turned true after long wear without encroaching on the taper.

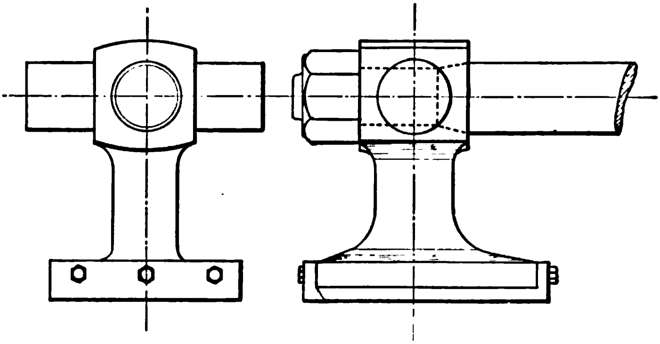

Fig. 48.—Piston-rod Crosshead.

It was usual to prolong the piston-rods of vertical engines so as to admit of the "tail" end passing through a stuffing-box in the cylinder-cover, and so help to guide the piston, and prevent its unduly wearing the cylinder. Since no moisture can get to the packing of this stuffing-box, and the lubricant applied externally soon gets carried through to the cylinder, some trouble is experienced in keeping it steam-tight; the rolling of the ship also causes the piston to exert pressure sideways on the gland and packing, and further aggravates the evil. For these reasons it is preferable to simply fit a brass or white metal bush in the cover for the "tail" end to work in, and cover it with a dome or sheath fitted steam-tight and true on the cover, making a couple of grooves in the side of the bush to admit and release the steam. But, on the whole, it is very doubtful if these tail-rods are so efficacious as represented, and they cannot be so beneficial to the good working of the cylinder as a piston with broad bearing surfaces. It should be noted that when the packing-ring is pressed out by springs acting independently of the body of the piston, it is advisable to form the piston with greater depth of flange and junk-ring.

The piston is secured to the rod by a nut, and the size of the rod should be such that the stress on the section at the bottom of the thread does not exceed 5500 lbs. per square inch for iron, and 7000 lbs. for steel. The depth of this nut need not exceed the diameter which would be found by allowing these stresses. To avoid the large cavity which is necessary in the cylinder-cover for the piston-rod nut, many engine builders recess it into the piston; this recess does not materially affect the strength of the piston, and the plan may be followed with advantage. Although piston-rod nuts seldom work loose, and those of vertical engines are less liable to this than are others, still as a measure of safety in all cases a taper-split pin should be fitted to the rod behind the nut, and in the case of large engines it is usual to fit a "lock" plate to the nut itself, or to adopt some other means of preventing it from moving at all when at work.

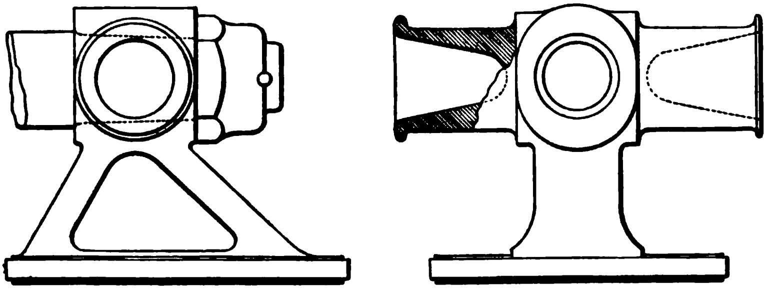

Fig. 49.—Cast-steel Crosshead.

Cast-steel Piston-rod Crosshead.—Fig. 49 represents the modern form of crosshead made of cast steel and designed in such a way as a casting permits of. It is, of course, much lighter and cheaper than that of forged material, shown in fig. 48, and this is especially so in fast running engines which require a large bearing surface. These crossheads can now be cast quite sound and of excellent material so as to be practically as good as a steel forging.

The stresses on the various sections are, as a rule, light, as the governing requirement of surface naturally causes it to be much larger than would be the case for mere strength. The gudgeons can be cast solid, or lightened out as shown. The example shown in fig. 49 has white metal fitted to both the head-going and stern-going sides, and is without a loose slipper. Such, however, can always be fitted to this type of crosshead if desired.

When cast steel is objected to as a material for the gudgeons, a composite design has been adopted whereby the crosshead is made of hard-wrought steel and fitted to the slipper, &c., and secured by studs and nuts.

Back-rods and Trunks.—Horizontal direct-acting engines had tail-ends to the piston-rods with slipper guides, working on a bracket secured to the cover, and stayed by rods so as to give a rigid bed to the slide. Return connecting-rod engines had usually a central back-rod secured by its flange to the piston, and, passing through a stuffing-box in the cylinder-cover, guided in the same way as above described (*vide* fig. 15). These rods were usually made of cast iron, and of much larger diameter than the piston-rods themselves, so that they added very materially in supporting the piston. The use of light steel pistons has obviated the necessity for these back-rods, and they are now never fitted.

Trunk Engines.—These engines, which have been described in Chap. i., have pistons with the trunks through which the connecting-rod passes, cast in one with them; the back-trunk, which serves as a means of getting at the gudgeon, as well as a guide or support to the piston, is bolted to the piston by a flange fitting into a recess, in the same way as the back-rod of the return connecting-rod engine. The crosshead or "gudgeon," as it is usually called, to which the connecting-rod is attached, is secured to two strong lugs well bracketed to the piston and front trunk by two bolts on either side, its ends being flattened and extended to take these bolts. These engines are arranged to turn so that the reaction or thrust of the connecting-rods is upward when going "ahead;" in this way the weight of the piston helps to balance the upward thrust, which otherwise would have a very prejudicial effect on the cylinder.

The thickness of the trunks at the junction with the piston is about the same as the thickness of the piston at that point, and this at the mouth about 0·7 of that at the root.

The diameter of the trunk depends on the length of stroke and diameter of connecting-rod at its middle: roughly

$$\text{Diameter of trunk} = 0{\cdot}75 \times \text{length of stroke.}$$

Piston-rod Guides.—The pressure on the piston is transmitted through the piston-rod to the connecting-rod, and the reaction of the latter rod acts in the direction of its length; consequently, when the connecting-rod is not in line with the piston-rod, the force of its reaction can be resolved into two component forces, one in the direction of the piston-rod, and the other perpendicular to it. This latter force is usually called the "thrust of the connecting-rod," and unless specially prevented, would tend to bend the piston-rod. To prevent such an occurrence, and to preserve the piston-rod in its true course, a guide is provided, and the piston-rod end fitted with blocks or slippers to work in it. This thrust varies from O at the end of the stroke to its maximum point, which is towards the point when the crank is at a right angle to the centre line through the cylinder, and depending on the cut-off point; when steam is cut off past half-stroke, then this is exactly the point of maximum thrust. To determine the magnitude of the thrust when

P is the total effective load on the piston, S the stroke, and L the length of connecting-rod, represented by A B, fig. 50:

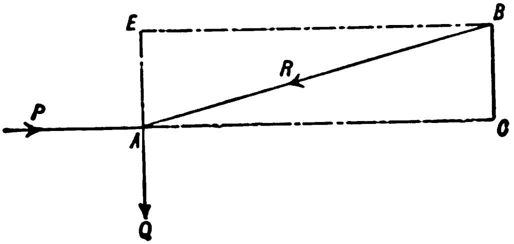

Fig. 50.

completing the parallelogram by the dotted lines A E, B E, A C, then the reaction, R, of the connecting-rod, represented in direction and magnitude by A B, will be resolved into two forces, P and Q, represented by the lines E B, A E, both in direction and magnitude. P must equal the load on the piston, and Q the thrust on the guides, may be obtained by measuring B C on a graphically constructed diagram, or by geometry calculated as follows:—

$$AC^2 = AB^2 - BC^2 = L^2 - \left(\frac{S}{2}\right)^2$$

Now,

$$P : Q :: AC : BC.$$

Therefore,

$$Q = P \times \frac{BC}{AC} = P \times \frac{\frac{S}{2}}{\sqrt{L^2 - \left(\frac{S}{2}\right)^2}} = P \times \frac{S}{\sqrt{4L^2 - S^2}}.$$

Or, by Trigonometry,

$$Q = R \text{ sine } BAC,$$

$$P = R \text{ cosine } BAC, \text{ or } R = \frac{P}{\text{cosine } BAC} = P \text{ secant } BAC.$$

Therefore,

$$Q = P\frac{\text{sine } BAC}{\cos BAC} = P \tan BAC.$$

The angle B A C is found by knowing its sine to be half the stroke ÷ length of connecting-rod.

Example.—To find the thrust taken on the piston-rod guide of an engine whose piston load is 100,000 lbs.; the length of stroke is 60 inches, and the connecting-rod is 120 inches long.

$$\text{Thrust} = 100,000 \times \frac{60}{\sqrt{4 \times (120)^2 - 60^2}} = 25,819 \text{ lbs.}$$

Surface of Guide-block.—The area of the guide-block, or slipper-surface on which the thrust is taken, when going *ahead* should be sufficiently large to prevent the maximum pressure exceeding 100 lbs. per square inch. When the surfaces are kept well lubricated this allowance may be exceeded, but the reduction in surface should be effected by making shallow grooves and recesses in the face of the slipper, in which the lubricant can lodge and impart itself to the guide as it is carried along. A good method of carrying this into effect is to provide a surface calculated on the allowance of 100 lbs. per square inch, and by cross planing so as to leave shallow recesses about $\frac{1}{16}$ inch deep, reduce the actual surface which touches the guide to about $\frac{5}{8}$ of the original area; there will then be strips across the slipper $1\frac{1}{4}$ inches wide, with depressions between them $\frac{3}{4}$ inch wide, filled with grease.

When the piston speed exceeds 300 feet per minute the area of guide-block should be increased to ensure continuous safe running. The safe pressure per square inch may therefore be taken at 1760 ÷ (piston speed in feet per minute + $100)^{\frac{1}{2}}$.

For example, an engine running at a piston speed of 900 feet per minute should have guide-blocks such that the maximum pressure per square inch does not exceed $\frac{1760}{\sqrt{900 + 100}}$, or 55 lbs. Guide-blocks designed on these lines for *ahead* motion have always ample provision for *astern-going*. Then in any engine

$$\text{Gross area of guide-block shoe} = \frac{T \times \sqrt{(S + 100)}}{1760} \text{ sq. ins.}$$

Cast iron, hard and close grained, is the best material for the guide-plates; its surface, after a few hours work, becomes exceedingly hard and highly polished, and offers very little resistance to the slipper or guide-block. So long as this hard skin remains intact, no trouble will be experienced, but if abrasion takes place from heating or other cause, it rarely works well after, and should be at once planed afresh.

The slippers or facing plates fitted to the piston-rod or crosshead are sometimes made of brass; but brass seldom gets that smooth hard skin so essential to good and efficient working, and when once the surface is grooved and scratched, it will wear away very rapidly. Cast iron is, after all, the best metal for this purpose, if care is taken at the first working of the engine to run for a few hours at easy speed, so as to rub down and polish the surfaces; after this is once thoroughly done, cast-iron surfaces will continue to work well with very slight attention. White metal is, however, often used for the facing of slippers, and works very well, and for high speed is essential for good and safe working. The best way of using white metal for this purpose is to fit strips of this material into grooves planed across a cast-iron slipper, and leave them standing from $\frac{1}{16}$ to $\frac{1}{8}$ inch above the cast iron. The strips should be about 2 inches wide, and the space between them about $1\frac{1}{4}$ inches, into which the lubricant can lodge, as before described. A slipper fitted in this way is shown in fig. 46. It is cheaper and more general to cast the white metal practically over the whole surface of the slipper, where it is retained in place by "tinning" and underlying the edges of the recess provided for it. It is usual to cut oil-ways in the face of the guide, which distribute the oil across it, and metal combs secured to the slipper dip into the oil receiver at the end of the guide, and smear the face on the return stroke.

The guide plates are sometimes planed across, instead of the slippers, for the same purpose of retaining the lubricant.

Piston-rod Crossheads and Gudgeons.—When there are two piston-rods, as in the case of the return connecting-rod engine, they are united to a common crosshead, having a turned journal in the middle for the connecting-rod to work on; or else a bearing is fitted to the crosshead, in which a gudgeon on the connecting-rod works. The former (fig. 51) is the better and usual plan under ordinary circumstances. This crosshead is of wrought-iron or steel, and made of a form suitable to the circumstances, and arranged to work in guides. The diameter at the middle must, of course, be sufficient to withstand the bending action, and generally from this cause ample surface is provided for good working; but in any case the area, calculated by multiplying the diameter of the journal by its length, should be such that the pressure does not exceed 1200 lbs. per square inch, taking the maximum *load on the piston* as the total pressure on it.

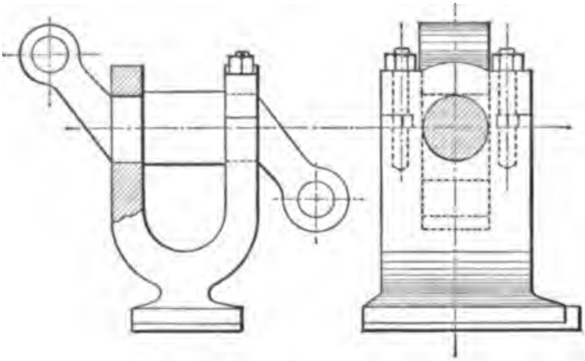

Fig. 51.—Crosshead and Guide-block.

Let L be the distance of the centres of the piston-rods in inches, and P the maximum load on the piston in pounds, then for strength

$$\text{Diameter of crosshead should not be less than } \frac{\sqrt[3]{P \times L}}{18}$$

For good wearing, l being the length of the journal,

$$\text{Diameter of crosshead should not be less than } \frac{P}{1200 \times l}.$$

With fast-running engines an allowance of 1200 lbs. is too high. The following rule, however, may be followed where P is the maximum load on the piston, d is the diameter and l the total length of gudgeon or crosshead arms in inches, and R the revolutions per minute.

$$d \times l = \frac{P \times \sqrt{R + 100}}{12{,}500}.$$

Then for any engine

$$\text{Diameter of crosshead} = \frac{P \times \sqrt{R + 100}}{12{,}500 \times l}.$$

Of course, the maximum load on the crosshead is really the reaction of the connecting-rod, but, to avoid any complication of the calculation, it is sufficient to take the load on the piston.

When the diameter of the crosshead journal is calculated by the first rule, the length is usually made equal to it.

Direct-acting engines have usually a gudgeon secured to the connecting-rod (fig. 52), which works in a bearing in the piston-rod end (figs. 45 and 46); or have the gudgeon at the piston-rod end (figs. 47, 48, and 49), and connecting-rod (figs. 53 and 53*a*) swung on it by brasses, &c., on either side. The advantage of the latter plan is the larger bearing surfaces obtainable, and the brasses being on the outside of the rods are much easier watched and adjusted; on the other hand, there are two sets of bolts, brasses, &c., to lubricate and keep in order, and there is the liability from carelessness to put the whole of the load on one side only. In the main, however, this plan is a preferable one for large and heavy rods, and it

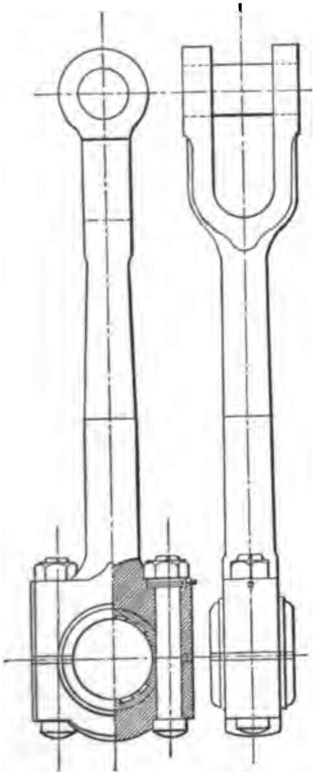

Fig. 52.

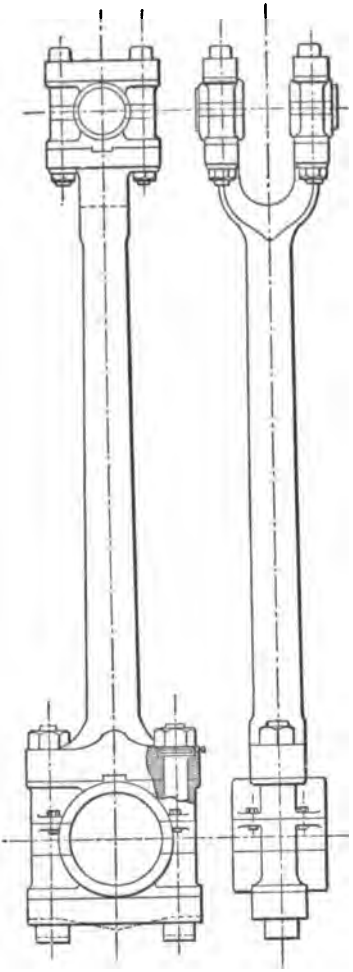

Fig. 53.
Connecting-rods.

is one which admits of the piston-rod being fitted into its end (figs. 48 and 49) instead of forged with it. This latter advantage is one well worthy of consideration, for it is highly important that the piston-rod shall be quite free from flaws and reeds on its surface, otherwise the packing soon gets damaged, and it is then found impossible to keep the glands from leaking. A steel rod rarely has them, but it cannot be so easily and cheaply forged with shoes, &c., on it; hence if the rod simply fits into the crosshead or rod end, this latter may with advantage be made of cast steel, while the rod is of forged mild steel.

Smaller engines are better with the gudgeon shrunk into the jaws of the connecting-rod, and working in brasses fitted into a recess in the piston-rod end (fig. 45), and secured by a wrought-iron cap and two bolts.

The diameter of the gudgeon = 1·25 × diameter of piston-rod.
„ length „ „ = 1·4 × „ „

The area obtained by multiplying these is exactly double the area of the piston-rod section, and so if the maximum stress per square inch on the rod does not exceed 2400 lbs., the above rule holds good; if this allowance of stress is exceeded in calculating the rod, then the *length* of the gudgeon should be increased until the pressure on the section, as calculated by multiplying length by diameter, does not exceed $\frac{12{,}500}{\sqrt{R + 100}}$ lbs. per sq. inch.

When the gudgeon is fixed in the piston-rod (fig. 47), or formed with the guide ends as in fig. 48, the length *of each* end should be not less than 0·75 the diameter of the piston-rod.

The brasses when fitted into the piston-rod should be square-bottomed and of hard gun-metal, with good oil-ways, as owing to the motion being through a comparatively small angle only, the lubricant is not so easily spread. White metal *does not* work so well on gudgeons and crossheads; it has a tendency to abrade, and to *wear the journal* oval. This is observed whenever white metal is used on a bearing subject only to a small angular motion, and is somewhat peculiar. White metal, however, should be used with mild steel gudgeons, especially in fast-running engines.

Gudgeons, when fitted to the connecting-rods, should be made of hard steel or wrought-iron case-hardened; great care is required, if of the latter, that they are carefully ground true after hardening.

The bolts securing the brasses should be of the best wrought iron, or mild steel; they should be of such a size that the stress per square inch of section *at the bottom of the thread* does not exceed 5000 lbs. when of iron, and 6500 when of steel. It is true that where extreme lightness of machinery is a *sine quâ non*, these stresses have been exceeded by some engineers; but even under these circumstances it is unwise to exceed them by more than 10 per cent., as the saving of weight effected is exceedingly small, especially when compared with the risk run (*v.* Table xlvi.).

When the bolts are less than 2 inches in diameter, owing to the uncertainty of the depth of thread, &c., the allowance should be 10 per cent. less than the above.

Diameter of piston-rod bolts at the bottom of thread

$$= D\sqrt{\frac{p}{f}}.$$

D is the diameter of piston, p, the effective pressure on it per square inch.

When there are two bolts of iron $f = 10{,}000$ lbs
" " " steel $f = 13{,}000$ lbs
" four " iron $f = 18{,}000$ lbs.
" " " steel $f = 24{,}500$ lbs.

The cap is of the same width as the piston-rod end—viz., 1·15 × diameter of the rod, and its thickness equal to the diameter of the bolts at *the bottom of the thread.*

Connecting-rods.—The length of the connecting-rod measured from the centre of the gudgeon to the centre of the crank-pin should, if possible, not be less than twice the stroke. Those of the horizontal direct-acting engines of warships can seldom be so long as this, and were generally one and three-quarter times the stroke; those of return connecting-rod engines are never less than twice the stroke, and sometimes as much as two and a half times; those of trunk engines two and a half to three times the stroke. Quick-running vertical engines have almost invariably the connecting-rods twice their stroke in length; other vertical engines from twice to three times, but generally two and a quarter times is not exceeded.

A connecting-rod may be viewed as a strut loose at both ends, the formula for which, as given on p. 163, is $R = \dfrac{f\,S}{1 + 4\,a\,\dfrac{l^2}{d^2}}$.

The value of R will be found by multiplying the load on the piston by the secant of the angle of obliquity of the connecting-rod (*vide* p. 168). Or by geometry

$$R = P \times \frac{2\,L}{\sqrt{4\,L^2 - S^2}},$$

P being the load on the piston, S the stroke, and L the length of the connecting-rod as before.

Simplifying the above formula by assuming a value r for the ratio of l to d, and substituting $\frac{\pi\,d^2}{4}$ for S; and taking the value of f at 3000 lbs.

$$\text{Diameter of connecting-rod in the middle} = \frac{\sqrt{R\,(1 + 4\,a\,r^2)}}{48{\cdot}5}.$$

Example.—To find the diameter at the middle of the connecting-rod of an engine, 60 inches stroke, whose length is 120 inches, and the load on the piston 100,000 lbs., r being taken at 15.

$$R = \frac{100{,}000 \times 2 \times 120}{\sqrt{4 \times 120^2 - 60^2}} = 103{,}358 \text{ lbs.}$$

$$\text{Diameter at middle} = \frac{\sqrt{103{,}358\left(1 + \dfrac{4 \times 15^2}{3000}\right)}}{48{\cdot}5} = 7{\cdot}6 \text{ inches.}$$

The following are the values of r in practice :—

Naval engines—Direct-acting - $r = 9$ to 11.
" " Return connecting-rod $r = 10$ to 13, old.
" " " " $r = 8$ to 9, modern.
" " Trunk - - - $r = 11{\cdot}5$ to 13.
Mercantile " —Direct-acting, ordinary $r = 12$.
" " " long stroke $r = 13$ to 16

Taking 10 as the average value of r for naval engines, and 13 for mercantile, then,

For a naval engine,

$$\text{Diameter of connecting-rod at middle} = \sqrt{\frac{R}{2000}}.$$

For mercantile engines,

$$\text{Diameter of connecting-rod at middle} = \sqrt{\frac{R}{1900}}.$$

The sizes given by these rules, although large enough for strength, are somewhat smaller than found in actual practice in the mercantile marine generally.

The following empirical formula will be found a very useful one, and the results given by it agree very closely with good modern practice:—

$$\text{Diameter of connecting-rod at middle} = \frac{\sqrt{L \times K}}{4}.$$

L is the length of the rod in inches, and

$$K = 0\cdot03 \times \sqrt{\text{Effective load on the piston in lbs.}}$$

Example.—To find the diameter of the connecting-rod, 100 inches long, for an engine having a load of 55,000 lbs.

$$K = 0\cdot03 \times \sqrt{55{,}000} = 7\cdot0,$$

$$\text{Diameter} = \frac{\sqrt{100 \times 7}}{4} = 6\cdot6 \text{ inches.}$$

The diameter of the connecting-rod at the ends may be 0·875 of its diameter in the middle. The tapering of rods, or making them barrel-shaped, is usual in the case of those having single brasses at both ends, such as are generally fitted to trunk and return connecting-rod engines; then the diameter of the crank-pin end is 0·925 of the diameter at middle. Direct-acting engines have usually the connecting-rods tapering from the gudgeon end to the middle, and then parallel, or nearly so, to the crank-pin end.

It is, however, simpler and sufficiently accurate to follow, in the case of connecting-rods, a similar rule to that laid down on p. 164 for piston-rods, viz.:—

$$\text{Diameter of connecting-rod} = \frac{\text{Diameter of cylinder}}{F} \sqrt{p}.$$

Here p is the boiler pressure absolute − 30 for the high-pressure cylinder of a triple or quadruple engine; one-half that pressure for the medium-pressure cylinder; and for the low-pressure cylinder that pressure divided by the ratio of low-pressure to high-pressure cylinder. F may be taken at 55 for the crosshead end of fast-running light engines, and 50 of mercantile engines of ordinary type. The diameter of the rods at the middle may be got by dividing by F − L, where L is the length of connecting-rod in feet.

Connecting-rod Bolts.—The diameter of the bolts may be calculated by allowing the same stress per square inch as that given for piston-rod bolts. It is usual now, from practical considerations, to make the bolts of both piston and connecting-rod of the same size; the bolts therefore should be calculated from the load on the connecting-rod. In order that the whole of the stretch shall not come on one section, as at the bottom of the last thread of an ordinary bolt, it is better to turn part of the body of connecting and piston-rod bolts to the same diameter as at the bottom of the thread, leaving it a little larger than the diameter over the thread close to the head, and in way of any joint—that is, the bolt is made with a *plus* thread, and bearing collars where required.

Connecting-rod Brasses.—The crank-pin brasses are more severely tried than any others about an engine, and, therefore, should be most carefully designed, and made of the very best material. Some engineers make the brasses to form the end of the rod (fig. 53), and retained to it by bolts and a wrought-iron cap; others prefer that they shall only act as bushes or liners to the connecting-rod, sometimes fitting them into a square or octagonal recess in the rod end, and held in place by a flat cap and bolts, just as is generally done to piston-rod ends; but more generally they are fitted in duplicate halves, as shown in fig. 52. The former plan is a very expensive one when they are of large size and made of brass, on account of the great weight required, and consequently are also costly to renew when worn, besides which they are very liable to get out of shape when heated, and to crack through the crowns. The latter plan avoids the use of so much brass, gives a good solid bed to the brasses, and leaves the bolts free of all strain except tension. When rods are made in this way it is usual to forge the head of the rod solid, and turn it and the cap at the same operation; the hole for the brasses is bored or slotted out (the latter when the hole is 9 inches and upwards in diameter) roughly; the head is then slotted through or parted in the lathe so as to cut off the cap, the space left by the tool being equal to twice the difference in thickness of the brass at the crown and sides; the cap is then bolted close to the rod, and the hole bored out to the diameter of the brasses measured across the rod. The brasses are kept from turning by a brass distance piece secured between the cap and rod and projecting between the brasses, and in the case of large brasses a short feather is fitted close to each flange in the crown. All brasses have a tendency to close on the pin or journal after having been hot, because the inner surface becomes warm first, and the metal in expanding tends to straighten the curved part; this is resisted by the other part of the brass and the bed in which it is fitted, and in consequence this inner surface gets compressed permanently, so that on cooling down it contracts, and tries then to give the brass more curvature, and so presses hard on the journal. This is the reason why some bearings will never work cool but always a trifle warm, this slight amount of heat causing the brass to expand so as to truly fit the journal. It is now not at all uncommon to make these fittings of cast iron when lined with white metal; indeed, cast iron carries white metal better than brass does, and when of good mixture is quite as strong as good gun-metal, and stronger than the common brass so often used with white metal.

Fig. 53*a* is an example of a modern connecting-rod with the jaws made as part of a cone, instead of as in fig. 53, thereby providing a better connection between each jaw and the body of the rod, as well as permitting of its being machined much more cheaply. This rod is fitted with cast-iron bearings at each end, lined with white metal. This form of rod and "brasses" is now very generally used in the mercantile marine for engines of all sizes.

White metal is better than bronze for the rubbing surface of the crank-pin "brasses," and it is important that the white metal shall project beyond the brass" so that it alone bears on the pin. For this purpose strips of white metal should be fitted into grooves planed in the "brass" and be well hammered, so as to thoroughly fill the spaces, after which it should be smoothly bored and fitted to the pin. Brasses which have not been originally designed for white metal may be fitted in this way, or by boring some shallow

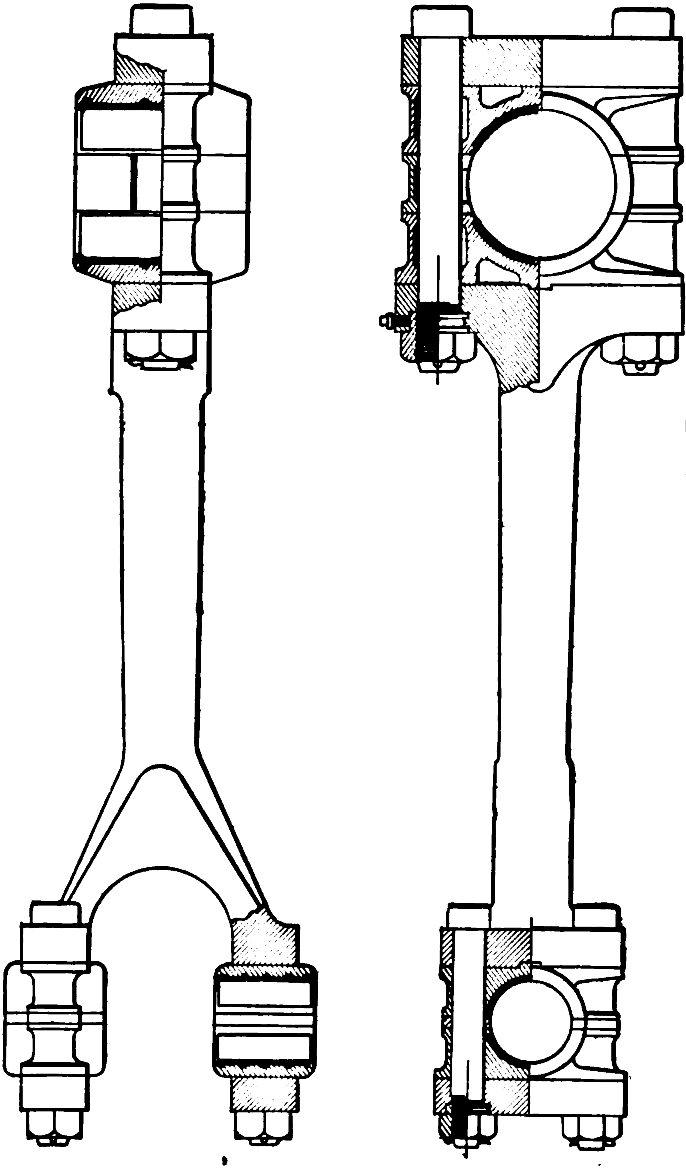

Fig. 53a.—Connecting-rod. Modern Form.

holes, whose diameter at the bottom is more than at the surface, casting into them buttons of white metal, and after hammering down boring out the brass so that the white metal stands out beyond the original wearing surface.

A very good plan, but somewhat more expensive and not more efficient than the one above described, is to run the white metal into recesses so cast with the brass, hammer it well in place, bore out, and then plane out the brass intervening between the white metal patches, leaving only slight ridges surrounding the latter, which prevent it from being spread out. The most general plan, however, is to have a recess in the "brass" or shell, which is now often of cast iron instead of bronze, tin the surface and run the metal in place while the shell is warm. When cold bore out as usual, trim the edges, &c., and cut grooves for the oil to circulate in the surface of the metal (*v.* fig. 53*a*).

Caps of Connecting-rod Brasses.—The width of the connecting-rod end should be a little larger than its diameter at the end; its thickness (in direction of the length) = 0·6 × diameter at middle.

The thickness of cap at middle = 0·8 × diameter of body of bolts + 0·1 × pitch of bolts.

Thickness of cap at ends = diameter of bolts at bottom of thread.

For ease of manufacture caps are generally made straight and of thickness given by the first rule.

Gudgeon End of Rod.—The jaws of every connecting-rod are subject to forces which tend to open them on the down stroke and close them on the up. When fitted with a gudgeon-pin secured in the eyes, as in fig. 52, the jaws are subject to bending moments at the various sections, which become maxima at a point just below the gudgeon and also at an angle of about 40° to the vertical through the centre of curvature of the inner portion. When the rod is fitted with "brasses" working on a crosshead, as in fig. 53, the maximum bending moment is at an angle of about 40° to the vertical or axis of the rod, and is greater than when with a gudgeon secured to the rod ends. When the diameter of the gudgeon or cross-head ends is

$$= \frac{\text{Diameter of cylinder}}{45} \times \sqrt{p},$$

then

Let P be the maximum re-current load on the piston.
L the width from centre to centre of gudgeon-rings or of crosshead brasses when so fitted.

$$F = P \div 3000, \text{ then}$$

Diameter of gudgeon-ring = diameter of gudgeon + 1·25 × F.
Thickness „ „ = 0·85 × F.
Width of jaw = 1·35 × F, or 1·1 × diameter of rod.
Thickness of jaw in line at 40° = ·48 F × L when with gudgeon.
„ „ „ = ·52 F × L when with brasses.

CHAPTER IX.

SHAFTING—CRANKS AND CRANK-SHAFTS, ETC.

MARINE shafting is invariably of circular section and generally solid, so that in speaking of a shaft one of this description will be always understood.

In practice, a shaft may be subject to a simple strain, such as is due to twisting or bending alone, or to the combined action of two or more strains. And, strictly speaking, every shaft is subject to bending and shearing strains caused by its own weight; but as these latter are usually so small, compared with the stresses to which shafts are subject from external forces, it is usual to omit them from consideration in all calculations of strength. Such strains, however, must sometimes come under consideration in very extreme cases, as when from some cause the bearings or supports are very far apart, or when a flywheel or other heavy fitting is attached to a shaft whose bearings are remote from the centre of gravity of the shaft and fittings.

Twisting Moment.—If a force is acting on a shaft so as to turn it, or tend to turn it, round on its axis, it is called a *twisting force*, and the *effort* of this force is measured by multiplying it by its distance from the axis, and called the *twisting moment* or *torque*. Suppose P is the thrust along the connecting-rod when at right angles to the crank, and L is the distance of the centre of the crank-pin from the centre of the shaft, P × L is the twisting moment on the shaft.

When *one* force is acting on the shaft as above described, the second force, which completes the *couple*, is the reaction of the bearing, which is equal to P, but acts in the opposite direction. If the force P and the reaction R act in a plane *perpendicular to the axis* of the shaft, they will cause no bending action on the shaft, but there will be a force R tending to shear the shaft across. But in actual practice it is almost impossible that P and R shall act in such a plane, and they usually act in planes parallel to one another, and perpendicular to the axis; hence, the shaft is also subject to a bending action. But if a shaft is turned by means of *two equal* forces acting in opposite directions, one on either side of the shaft and equidistant from the axis and in the same plane, then the shaft is balanced, these forces will cause no pressure on the bearings, and it is subject therefore to twisting strains only. If one shaft is coupled to another shaft, from which it is to transmit power by two coupling bolts equidistant from the centre, it will only receive a twisting strain. Such is the state of the shafting from the crank-shaft to the propeller-shaft of a screw steamship.

Resistance to Twisting.—Let T be the twisting moment on a shaft in inch pounds, d the diameter of the shaft in inches, and f the stress per square inch on the transverse section of the shaft. Then (Rankine, *Applied Mechanics*, p. 355)

$$T = \frac{\pi d^3}{16} f; \quad \text{or } d = \sqrt[3]{\frac{T}{f} \times 5{\cdot}1}.$$

For wrought iron f should not exceed 8000 lbs., and when the shafts are more than 10 inches diameter 7000 lbs. is the highest stress to which the iron should be subjected. Steel, when made from the ingot and of good materials, will admit of a stress of 10,500 lbs. for small shafts, and 9000 lbs. for those above 10 inches diameter. The difference in the allowance between large and small shafts is to compensate for the defective material observable in the heart of large shafting, owing to the hammering failing to perfect it. In practice, shafts for continuous work are designed for stresses 20 to 35 per cent. less than those here given to allow for strains beyond those included in the calculations, and permit of longer service. When, however, the work is not continuous and the full power only occasionally employed, even higher allowances may be made for f (*v.* Table xlvi.), but always bearing in mind the fact that resistance to shear is only 80 per cent. of the tensile strength of wrought iron and mild steel.

Example.—To find the diameter of an iron shaft subject to twisting only, the force being 100,000 lbs. acting at a distance of 24 inches.

$$T = 100{,}000 \times 24 = 2{,}400{,}000 \text{ inch-lbs.}$$

$$d = \sqrt[3]{\frac{2{,}400{,}000}{8000} \times 5{\cdot}1} = 11{\cdot}5 \text{ inches.}$$

Diameter of a Shaft subject to Torsion.—If a constant force P were applied to the crank-pin tangentially to its path, then the work done per revolution will be $P \times \frac{2\pi L}{12}$; L being the length of the crank in inches; then if R be the number of revolutions per minute,

$$\text{Work done per minute} = P \times \frac{2\pi L}{12} \times R. \quad - \quad - \quad (1)$$

But this work is equal to I.H.P. × 33,000; and the twisting moment is P × L constantly. Then

$$(P \times L) \times \frac{2\pi}{12} \times R = \text{I.H.P.} \times 33{,}000,$$

and

$$P \times L = \frac{\text{I.H.P.} \times 33{,}000 \times 12}{2\pi \times R}.$$

That is,

$$\text{Mean twisting moment} = \frac{\text{I.H.P.}}{R} \times 63{,}000. \quad - \quad - \quad - \quad - \quad (2)$$

And as before

$$d = \sqrt[3]{\frac{\text{I.H.P.} \times 63{,}000 \times 5{\cdot}1}{R \times f}}$$

$$= \sqrt[3]{\frac{\text{I.H.P.}}{R} \times \frac{321{,}300}{f}}. \quad - \quad - \quad - \quad - \quad (3)$$

If f be taken at 7500 for iron

$$\text{Diameter of shaft} = \sqrt[3]{\frac{\text{I.H.P.}}{R} \times 42{\cdot}84}. \quad - \quad - \quad - \quad - \quad (4)$$

But as shafts must be strong enough to resist the *maximum* twisting stress, it is necessary always to base calculations on it instead of on the mean twisting moment. The factor 42·84 must, therefore, be multiplied by the ratio of maximum to mean moment, as given in Table xvi.

Professor Rankine directs (*Rules and Tables*, p. 250), in order to find the greatest twisting moment from the mean: if a shaft is driven by a single engine, multiply by 1·6; if by a pair of engines with cranks at right angles, by 1·1; if by three engines with cranks at angles of one-third of a revolution, by 1·05.

These values are, however, very much lower than usually obtained in practice.

For a three-crank engine, cranks at 120°, cutting off at half to three-quarter stroke, multiply by 1·15.

For a two- and four-crank engine having cranks at right angles, cutting off steam at half to three-quarter stroke, multiply by 1·3.

For a single-cylinder engine cutting off steam at half stroke, by 2·0.

The following rule holds good for the ordinary engines, as found in general use in the merchant service:—

$$\text{Diameter of the tunnel shafts} = \sqrt[3]{\frac{\text{I.H.P.}}{\text{R}}} \times \text{F}. \qquad (5)$$

Compound engines, cranks at right angles—

Boiler pressure 70 lbs., rate of expansion 6 to 7, F = 70.
„ 90 „ „ 7 to 9, F = 74.
„ 110 „ „ 8 to 10, F = 78.

Triple-compound, three-cranks at 120° —

Boiler pressure, 150 lbs., rate of expansion 10 to 12, F = 62.
„ 165 „ „ 11 to 13, F = 64.
„ 180 „ „ 12 to 15, F = 67.
„ 200 „ „ 13 to 16, F = 70.

Quadruple-compound engines, two cranks, right angles, F = 72.
„ „ four cranks, F = 70.

Expansive engines, cranks at right angles, and the rate of expansion 5, boiler pressure 60 lbs., F = 90.

Single-crank compound engines, pressure 80 lbs., F = 96.

The shafts of torpedo-boats, destroyers, and fast yachts which are run at full speed only occasionally, and for short periods, may be designed by taking F at about a half of the above values.

Bending Moment.—If a force is acting on a shaft tending to bend it only, its effort is called the *bending moment*, and is measured by multiplying the force by the distance at which it acts from the support of the shaft.

If the shaft is overhung like a cantilever, and a force P is applied at a distance L from the point of support,

$$\text{The bending moment} = \text{P} \times \text{L} \qquad (1)$$

If supported on two bearings, whose distance apart is L, and a force P is applied at a point *midway* between these two bearings,

$$\text{The bending moment} = \frac{\text{P} \times \text{L}}{4}. \qquad (2)$$

If the bearings are long—that is, exceeding the diameter of the shaft in length, and are also strong and rigid, so that the shaft is *held* by them sufficiently to prevent flexure taking place in the bearing,

$$\text{The bending moment} = \frac{\text{P} \times \text{L}}{8}. \qquad (3)$$

Since, however, few shafts are so secured as to comply with these conditions exactly, any shaft supported in strong bearings not less than 1 diameter long, and whose distance apart does not exceed 10 diameters, and which has to work freely in its bearings, may be treated as partly complying with these conditions, and

$$\text{The bending moment} = \frac{P \times L}{6}. \quad - \quad - \quad (4)$$

When the bearings are not rigid enough to prevent flexure of the shaft within them, L must be measured from the centres of the cap bolts, so that where each cap is held down by a pair of bolts L is measured to the centres of bearings. If the caps and bearings are strong and rigid enough to resist any tendency to bend by the action of the shaft, L may be measured from the edge of the bearing or cap. If the bearing is fitted with brasses, which project beyond the cap and bed so much as to receive little or no support from them, L must still be measured from the edge of cap. In a few words, the distance must be measured from what would be the actual points of support if it is bent by severe pressure.

Resistance to Bending.—The strength of a circular section shaft to resist bending is only half of that to resist twisting. If M is the bending moment in inch pounds, and d the diameter of the shaft in inches,

$$M = \frac{\pi d^3}{32} \times f; \quad \text{and} \quad = \sqrt[3]{\frac{M}{f} \times 10{\cdot}2}$$

f is a factor which depends on the material of which it is composed, and the value may be based on the fact that it is in tension and compression. The only shafts in a marine engine which are subject to *bending only* are some weigh-shafts having double-ended levers, similar to the side levers of paddle-wheel engines, and their diameter is determined from other considerations than that of mere strength; but with them, as with the crossheads of return connecting-rod engines, care should always be taken that the size suitable for good working in the bearings should be sufficient for strength.

Equivalent Twisting Moment.—When a shaft is subject to both twisting and bending simultaneously, the combined stress on any section of it may be measured by calculating what is called the *equivalent twisting moment*—that is, the two stresses are so combined as to be treated as a twisting stress only of the same magnitude, and the size of shaft calculated accordingly. Professor Rankine gave the following solution of the combined action of the two stresses (*vide* Rankine, *Rules and Tables*, p. 227):—Let T be the twisting moment on a shaft when M is the bending moment on a section, then taking T_1 as the equivalent twisting moment.

$$T_1 = M + \sqrt{M^2 + T^2}.$$

Example.—To find the diameter of a section of a shaft at which the bending moment is 40,000 inch-pounds, when the twisting moment is 250,000 inch-pounds. The shaft of iron $f = 7500$ lbs.

Here

$$\begin{aligned} T_1 &= 40{,}000 + \sqrt{40{,}000^2 + 250{,}000^2} \\ &= 40{,}000 + 10{,}000\sqrt{4^2 + 25^2} \\ &= 293{,}170 \text{ inch-pounds.} \end{aligned}$$

$$d = \sqrt[3]{\frac{T}{f} \times 5{\cdot}1} = \sqrt[3]{\frac{293{,}170 \times 5{\cdot}1}{7500}} = 5{\cdot}84 \text{ ins}$$

Crank Shafts.—These shafts are subject always to twisting, bending, and shearing stresses; the latter are so small compared with the former that they are usually neglected directly, but allowed for indirectly by means of the factor f, as already stated.

The two principal stresses vary throughout the revolution, and the maximum equivalent twisting moment can only be obtained accurately by a series of calculations of bending and twisting moments taken at fixed intervals, and from them construct a curve of strains.

Curve of Twisting Moments.—The twisting moment at any position of the crank is equal to the pressure on the piston multiplied by the distance intercepted by a line through the connecting-rod on a line at right angles to centre line through centre of cylinder.

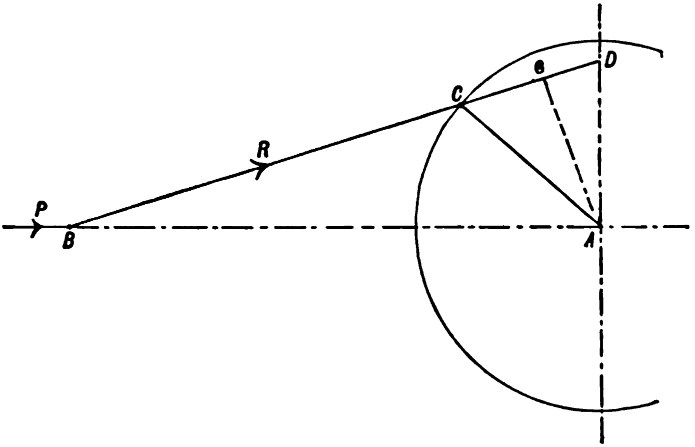

Fig. 54.

Let A B (fig. 54) be the centre line of the engine through the cylinder and shaft centres, A C the position of the crank, B C the connecting-rod, and A D a line at right angles to A B. Produce B C to cut the line A D, and drop from A a line A E perpendicular to B C. P is the load on the piston, and R is the thrust on the connecting-rod. It will easily be proved that the angle D A E is equal to the angle A B D, called for convenience α. Then

$$P = R \cos a; \text{ and } AE = AD \cos \alpha.$$

$$\text{The twisting moment} = R \times AE = R \times AD \cos \alpha = P \times AD.$$

Let the twisting moment be calculated at equal intervals of say 10° of angular movement of the crank, so that in the whole revolution there will be 36 observations, or 18 in the half revolution. Draw a line A B (fig. 55), and divide it into 18 equal parts, Aa_1, a_1a_2, &c.; erect at these points perpendiculars, and cut off parts a_1b_1, a_2b_2, &c., to represent the value of the twisting moments at each corresponding position of the crank to a suitable scale. Through the points b_1 b_2, &c., draw a curve, which represents the curve of strain on the shaft during the forward movement of the piston; by producing A B, and going through a similar operation for the second half of the revolution, the curve of strain during the backward movement of the piston can be obtained.

Divide the area enclosed between this curve and the line A B by the length of A B, and the quotient is the *mean* twisting moment, and represented by A M in fig. 55, so that the rectangle A M N B is equal in area to the figure A B C.

The value of A M may be calculated by taking a mean of the values of a_1b_1, a_2b_2, &c.

When there are two engines, that is, two pistons operating on one shaft, the combined twisting moment is found by drawing the curve of twisting moments of each crank separately, transposing that of one on that of the other in a position corresponding to the relative position of the cranks. In fig. 56 A C B is the curve of strain on one crank, and $A_1C_1B_1$ that on the other, which is at an angle with it of degrees represented by AA_1. The combined twisting moment at any period *a* is represented by *a d*, which is equal to *a b* + *a c*, and the dotted curve CdC_1, &c., represents the curve of combined twisting moments.

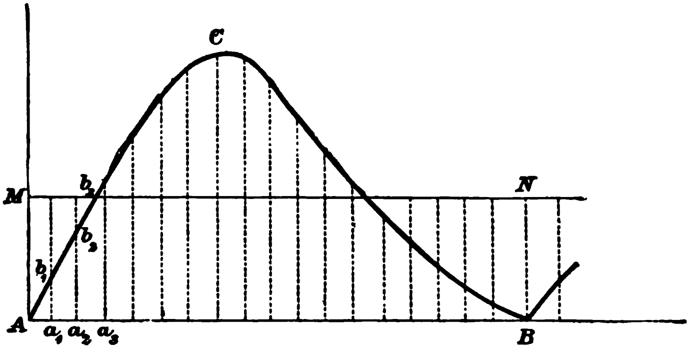

Fig. 55.—Curve of Twisting Moments.

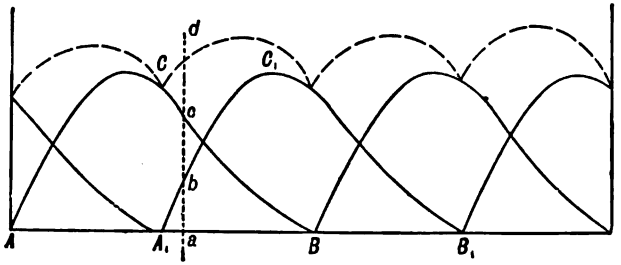

Fig. 56.—Curve of Combined Twisting Moments.

The maximum twisting moment will be at the point where the curve is highest, and the ordinate may be measured and its value found by referring to the scale to which the curve is drawn. The mean twisting moment may be found by measuring the area included between the dotted curve and the base line and terminal ordinates, and dividing by the base line, or by taking a mean value from the ordinates as before.

If there are three engines, a similar operation will indicate the maximum twisting moment.

The pressures at the different points may be taken from the indicator-diagrams, or by constructing diagrams from the conditions under which the engine is to work.

The bending moment on a section of the shaft will vary exactly with the pressure on the crank-pin, and to find the maximum equivalent twisting moment on a section, it is only necessary to construct a secondary curve from the formula $T_1 = M + \sqrt{M^2 + T^2}$ between the point of maximum twisting and that at which the pressure on the piston is greatest.

When steam is not cut off in the cylinder before 0·4 of the stroke, the maximum load on the piston may be used to calculate the bending moment, which is to be combined with the maximum twisting moment to find the maximum equivalent twisting moment. Only when steam is cut off earlier than this does the point of maximum equivalent twisting moment differ much from the point of maximum twisting.

Momentum of Moving Parts.—In making these calculations it has been assumed that the moving parts, such as the piston and rods, have no effect on the force exerted on the shaft; but this is not strictly true, for since these parts are of considerable weight, a part of the energy of the steam is absorbed at the commencement of the stroke in overcoming their inertia, and consequently the strain on the crank-pin is less then than is represented on the curves. Again, towards the end of the stroke the *momentum*, or energy thus stored in these moving parts, is given out on the crank-pin, and causes larger strains on it than shown by the curve. The further consideration of the effect of the inertia and momentum of the moving parts will be found in Chap. xxiv. It is sufficient to say here that the general tendency is to modify the stresses on the crank shaft, as also those of the connecting-rods, piston-rods, &c., so that if any of these are strong enough to withstand the stresses due to external forces, they are sufficiently strong for the engines when moving. Indeed, it is only at very high speeds that momentum need be taken into account.

The following Table (xvi.) gives the relation between the maximum and mean twisting moments of engines working under various conditions, the momentum of the moving parts being neglected:—

TABLE XVI.

Description of Engine.	Steam cut-off at	Max. Twist. / Mean Twist.
Single-crank,	0·2	2·625
,,	0·4	2·125
,,	0·6	1·835
,,	0·8	1·698
Two-cylinder, cranks at 90°,	0·1	1·872
,, ,,	0·2	1·616
,, ,,	0·3	1·415
,, ,,	0·4	1·298
,, ,,	0·5	1·256
,, ,,	0·6	1·270
,, ,,	0·7	1·329
,, ,,	0·8	1·357
Three-cylinder compound, three cranks at 120°,	H.P. 0·5, L.P. 0·66	1·40
,, triple, ,, ,,	0·6	1·15
Four-cylinder quadruple, four cranks at 90°,	0·6	1·26

Overhung Crank.—The simplest form of crank is that known as the overhung crank, such as is usually fitted in mill engines, but hardly found now in marine engines. The shaft projects beyond the bearing, and has keyed to its end a lever or disc, in which is secured the crank-pin.

The pin is subject to bending and shearing forces, due to the thrust on the connecting-rod. The maximum bending moment on the part of the pin close to the crank is found by multiplying the greatest thrust of connecting-rod by the distance to the centre of the connecting-rod.

If R is the thrust of the connecting-rod, and l the length of the pin, then

$$\text{* Bending moment on crank-pin} = \frac{R \times l}{2},$$

$$\text{and diameter of pin} = \sqrt[3]{\frac{R \times l}{2} \times \frac{10{\cdot}2}{f}} = \sqrt[3]{\frac{R \times l}{f} \times 5{\cdot}1}.$$

Example.—To find the diameter of the crank-pin whose length is 14 ins. and the thrust of connecting-rod is 125,000 lbs., f being of steel and taken at 9000 lbs.

$$\text{Diameter} = \sqrt[3]{\frac{125{,}000 \times 14}{9000} \times 5{\cdot}1} = 9{\cdot}97 \text{ ins.}$$

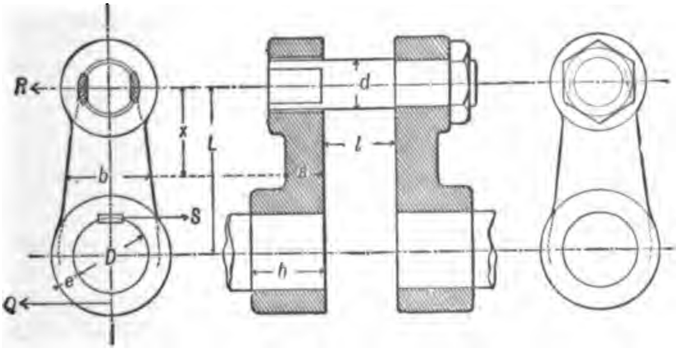

Fig. 57.—Cranks of Paddle-wheel Engine.

The crank-arm (fig. 57) is to be treated as a lever, so that if a is the thickness in direction parallel to the shaft axis, and b its breadth at a section x inches from the crank-pin centre, then

$$\text{Bending moment M at that section} = R \times x,$$

and

$$\frac{a \times b^2}{6} = \frac{M}{f},$$

or

$$a = \frac{6\,M}{b^2 \times f}.$$

If a crank-arm were constructed so that b varied as $\sqrt{x}$ (as given by the above rule), it would be of such a form as to be inconvenient of manufacture, and consequently it is customary in practice to find the maximum value of b, and draw tangent lines to the curves at the points; these lines are generally, for the same reason, tangential to the boss of the crank-arm at the shaft.

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* For other conditions as to size of pins, *vide* p. 195.

The bending moment decreases as the distance from the crank-pin decreases, while the shearing stress is the same throughout the crank-arm; consequently this latter stress is large compared with the bending stress close to the crank-pin, and so it is not sufficient to provide there only for bending stresses. The section at this point should be such that, in addition to what is given by the calculation from the bending moment, there is an extra square inch for every 8000 lbs. of thrust, on the connecting-rod. Moreover, the crank arm is subject to twisting from the action of the pin; strictly speaking, therefore, it should be calculated from the formula $T_1 = M + \sqrt{M^2 + T^2}$.

The length of the boss *h*, into which the shaft is fitted, is from 0·75 to 1·0 of the diameter of the shaft, and its thickness *e* must be calculated from the twisting stress R × L.

The crank turns the shaft (fig. 57) by exerting a force S on the key, whose centre of effort is on the circumference, and therefore at a distance of half the diameter from the axis of the shaft, so that

$$S \times \frac{D}{2} = R \times L; \qquad \text{or } S = 2\,R \times \frac{L}{D}.$$

If the crank is loose, the area of the section of the key parallel to the shaft must therefore not be less than S ÷ 10,000 lbs. And the strain on the section of the crank-boss opposite the key is

$$Q = S - R = R\left(\frac{2\,L}{D} - 1\right).$$

The stress on the section of the boss crossways is T, so that

$$T\frac{D + e}{2} = R \times L; \qquad \text{or } T = 2\,R\frac{L}{D + e}.$$

The stress on this section should not exceed 8000 lbs. if of iron, and 9500 if of steel.

To avoid a complicated expression, it is convenient to assume a relation between *h* and *e*, and to substitute the value of *e* thus found in the above expression. The value of $\frac{h}{e}$ in practice varies from 2, when there is not much space for the crank, to 3, when there is ample room.

Example.—To find the section of the boss of a wrought-iron crank 8 inches long; the pressure on the crank-pin is 54,000 lbs., the diameter of the shaft 10 inches, and $\frac{h}{e}$ assumed at 2·2. Stroke of piston 60 inches.

Here assume

$$e = \frac{8}{2{\cdot}2} = 3{\cdot}67 \text{ inches.}$$

$$T = 2 \times 54{,}000\,\frac{30}{10 + 3{\cdot}67} = 237{,}015 \text{ lbs.}$$

$$\text{Area} = 237{,}015 \div 8000 = 29{\cdot}63 \text{ square inches.}$$

And since

$$h = 8 \text{ inches, } e = \frac{29{\cdot}63}{8} = 3{\cdot}704 \text{ inches.}$$

The cranks of marine engines are always of steel or wrought iron, and generally of the same materials of which the shaft is made, so that the length and thickness of boss may bear a constant relation to the diameter of the shaft.

When $h = D$, then $e = 0{\cdot}35$ D, if steel, $0{\cdot}30$ D.
„ $h = 0{\cdot}9$ D, „ $e = 0{\cdot}38$ D, „ $0{\cdot}32$ D.
„ $h = 0{\cdot}8$ D, „ $e = 0{\cdot}40$ D, „ $0{\cdot}33$ D.
„ $h = 0{\cdot}7$ D, „ $e = 0{\cdot}41$ D, „ $0{\cdot}34$ D.

The crank-eye or boss into which the pin is fitted should bear the same relation to the pin that the boss does to the shaft.

Cranks are always shrunk on to both shaft and pin, and when this operation is carefully and well done, a key to the latter is almost unnecessary, and some engineers have latterly omitted to fit one to even very large pins; some engineers simply drill a hole half into the shaft and half into the crank, and drive into it a steel pin so as to answer the purpose of a key.

The diameter of the shaft end on to which the crank is fitted should be 1·1 × diameter of the journal. Overhung cranks are never fitted now to screw engines, as they often proved to be very unsatisfactory, from the fact of the whole of the pressure coming on one bearing, and the whole of the bending and twisting stresses being taken by one crank and journal.

Paddle-shafts.—The cranks of a paddle-wheel engine (fig. 57) are still often overhung, and in the case of double engines, the arm to which the pin is secured is the one fitted to the intermediate shaft; the pin fits loosely into an eye on a crank or disc secured to the paddle-shaft, and so drives this latter shaft. The effect of this arrangement is to give a very equable strain to the paddle-shaft, for the pressure of the pin is always at right angles to the crank on the paddle-shaft; and in smooth water the power of each engine is very nearly equally divided between the two wheels, and the *bending* action on the *paddle*-shaft never exceeds half that due to its own cylinder, for when near the dead points the bending moment is at its maximum, and is wholly taken on the crank-arm to which the pin is secured. For these reasons the shaft to which the arm having the crank-pin secured is fitted must be stronger than the outer shafts, especially when the ship is intended to work in rough water, as it is *liable* then to have to transmit the *whole* twisting force of one engine, and *always* takes, during certain periods of the revolution, the whole bending force from that engine. Hence, if T be the maximum twisting moment from one piston of a double paddle-wheel engine, and M the maximum bending moment from that piston, the

Maximum equivalent twisting moment on the intermediate shaft

$$= M + \sqrt{M^2 + T^2},$$

And maximum equivalent twisting moment on the paddle-shaft

$$= \frac{M}{2} + \sqrt{\left(\frac{M}{2}\right)^2 + T^2}.$$

Exception may be taken to the latter, since at times when one wheel is out of water the whole of the twisting force of both engines is transmitted through the shaft of the wheel which is deeply immersed; but when the maximum combined effect of twisting is on this one shaft, the bending moment on the crank-journal is probably less than $\frac{M}{2}$, and is that due to the force found by dividing the maximum twisting moment by the length of the crank, which is approximately $\frac{T \times \sqrt{2}}{L}$; the distance at which this force acts is measured from the face of the crank-arm to the edge of the casting, into which the journal brass is fitted.

Example.—To find the diameter of intermediate and paddle-shafts of a double paddle-wheel engine, having cylinders 80 inches diameter and 60 inches stroke, using steam of 45 lbs. pressure absolute, cutting off at 0·6 the stroke; the distance between the bearing beds being 50 inches.

Maximum effective pressure in the cylinder will be about 40 lbs. per square inch. Hence

$$\text{Load on piston} \quad - \quad - \quad = 5026 \times 40, \text{ or } 201{,}040 \text{ lbs.}$$

$$\text{Maximum twisting moment} = 201{,}040 \times 30 = 6{,}031{,}200 \text{ inch-lbs.}$$

$$\text{Maximum bending moment} = \frac{201{,}040 \times 50}{4} = 2{,}513{,}000 \text{ inch-lbs.}$$

(1) Maximum equivalent twisting moment on intermediate shaft

$$= 2{,}513{,}000 + \sqrt{2{,}513{,}000^2 + 6{,}031{,}200^2} = 9{,}056{,}000 \text{ inch-lbs.}$$

Diameter of shaft

$$= \sqrt[3]{\frac{9{,}056{,}000}{8000}} \times 5{\cdot}1 = 17{\cdot}9 \text{ inches.}$$

(2) Maximum equivalent twisting moment on paddle-shaft

$$= \frac{2{,}513{,}000}{2} = \sqrt{\left(\frac{2{,}513{,}000}{2}\right)^2 + 6{,}031{,}200^2} = 7{,}417{,}500 \text{ inch-lbs.}$$

Diameter of shaft

$$= \sqrt[3]{\frac{7{,}417{,}500}{8000}} + 5{\cdot}1 = 16{\cdot}8 \text{ inches.}$$

The outer part of a paddle-wheel shaft is subject to twisting and bending from the reaction of the water on the floats, and from bending due to the weight of the wheel itself. The pressure on the float can be found by dividing the twisting moment by the distance to the centre of the pressure of the float from the shaft axis in inches. It is practically sufficiently accurate to measure to the centre of fixed floats, and to gudgeons of feathering floats.

For example, the reaction of the water on the floats of the engine in the preceding example, whose radius to float centres is 140 inches, will be found

$$\text{Pressure on floats} = \frac{6{,}031{,}200}{140}, \text{ or } 43{,}080 \text{ lbs.}$$

The twisting moment on the shaft is the same at the outer bearing as at the inner, and is 6,031,200 inch-lbs. The weight of the wheel is 20 tons, or 44,800 lbs., and the distance of its centre from the bearing is 40 inches.

* Maximum bending moment

$$= (44{,}800 + 43{,}080) \times 40 \text{ inches} = 3{,}515{,}200 \text{ inch-lbs.}$$

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* In smooth water the bending force is really the *resultant* of the weight and reaction on the floats, and may be taken $= \sqrt{\text{weight}^2 + \text{reaction}^2}$.

Maximum equivalent twisting moment

$$= 3{,}515{,}200 + \sqrt{3{,}515{,}200^2 + 6{,}031{,}200^2} = 10{,}515{,}200.$$

Diameter of shaft

$$= \sqrt[3]{\frac{10{,}515{,}200}{8000}} \times 5{\cdot}1 = 18{\cdot}88 \text{ inches.}$$

The outer end of a paddle shaft is subject to alternating stresses due to the weight of wheel, and the inner part to intermittent ones (*v.* Table xlvi.).

The crank-shafts of paddle engines are now made in the same way as those screw engines. Sometimes they are made in one piece from a "solid" forging, and sometimes in one piece "built up." Those in very large engines have a separate shaft for each cylinder, coupled as in screw engines (*vide* fig. 8).

Crank-shaft of Screw Engines.—In case of the forward crank of a double or treble engine, and the crank of a single engine having two arms, there is the action of one engine only on it. On the *forward* journal and crank-arm there is a twisting action sufficient to overcome the friction, and to drive the eccentrics if fixed in this part, and half of the whole bending moment due to the thrust on the crank-pin. On the *aftward* journal, the other half of the bending moment, and the whole of the twisting moment, except the small portion required as above; this portion is at certain periods of the revolutions so small, that in calculations for the journals it may be neglected.

Then equivalent twisting moment on aftward journal

$$= \frac{M}{2} + \sqrt{\left(\frac{M}{2}\right)^2 + T_2}.$$

Strain on forward journal $= \dfrac{M}{2}$.

In multiple-crank engines the aftward crank has not only to resist the action of its own piston, but also to transmit the twisting strains of the forward engines. There will be strains from its own piston, which may be calculated in the same way as those on the forward cranks, and to these must be added the twisting strain of the forward engine.

Let T_2 be the maximum twisting moment on the after engine from its own piston, and M_2 the corresponding bending moment, T_1 the twisting moment on the forward engines at the same period.

Then on the *forward* journal of the after crank, the twisting moment is T_1, and the bending strain $\dfrac{M_2}{2}$, so that—

Equivalent twisting moment on forward journal of after crank

$$= \frac{M_2}{2} + \sqrt{\left(\frac{M_2}{2}\right)^2 + T_1^2}.$$

On the *after* journal of the aftward crank, the twisting moment is $T_2 + T_1$, and the bending moment $\dfrac{M_2}{2}$, so that—

Equivalent twisting moment on after journal of aftward crank

$$= \frac{M_2}{2} + \sqrt{\left(\frac{M_2}{2}\right)^2 + (T_2 + T_1)^2}.$$

The bending moment on the after-arm of the aftward crank will be found by calculating the *maximum force on the crank-pin tending to twist the shaft.*

Let T_m be the maximum combined twisting moment, as found by the methods indicated before, L the length of the crank or half-stroke of piston. Then the maximum twisting force at the crank-pin is $T_m \div L$.

The maximum bending moment at any section of the after crank-arm of the aftward crank, whose distance from the centre of the crank-pin is x inches, is $\frac{T_m}{L} \times x$.

The maximum bending moment on a section of the forward arm of the same crank is $\frac{T_1}{L} \times x$.

Example.—To find the sizes of the parts of a crank-shaft of a double expansive engine of 1000 I.H.P., the length of stroke is 40 inches, the cut-off 0·6, and the stroke and the cranks at right angles. Revolutions 60 per minute. Mean twisting moment of one engine $= \frac{500}{60} \times 63{,}000$. Since the cut-off is 0·6, the ratio of maximum to mean twisting moment is 1·835 (Table xvi.); therefore

Maximum twisting moment of one engine

$$= 1{\cdot}835 \times \frac{500}{60} \times 63{,}000 = 963{,}375 \text{ inch-lbs.}$$

Mean twisting moment of both engines

$$= \frac{1000}{60} \times 63{,}000.$$

Ratio of maximum to mean twisting moments is 1·27 (Table xvi.); therefore

Maximum twisting moment of both engines

$$= 1{\cdot}27 \times \frac{1000}{60} \times 63{,}000 = 1{,}333{,}500 \text{ inch-lbs}$$

Maximum turning force on forward pin

$$= \frac{963{,}375}{20} = 48{,}168 \text{ lbs.}$$

Maximum turning force on aftward pin

$$= \frac{1{,}333{,}500}{20} = 66{,}675 \text{ lbs.}$$

Assuming the distance between the bearings on which the brasses are bedded to be 30 inches,

The maximum bending moment on each of the two forward journals

$$= \frac{48{,}168 \times 30}{8} = 180{,}630 \text{ inch-lbs.}$$

That on the two journals of aftward crank

$$= \frac{66{,}675 \times 30}{8} = 250{,}000 \text{ inch-lbs.}$$

Then diameter of foremost journal

$$= \sqrt[3]{\frac{180,630}{8000}} \times 10{\cdot}2 = 6{\cdot}13 \text{ inches.}$$

The maximum equivalent twisting moment on after journal of forward crank

$$= 180,630 + \sqrt{180,630^2 + 963,375^2} = 1,160,630 \text{ inch-lbs.}$$

Diameter of journal

$$= \sqrt{\frac{1,160,630}{8000}} \times 5{\cdot}1 = 9{\cdot}04 \text{ inches.}$$

The maximum equivalent twisting moment on fore journal of aftward crank

$$= 250,000 + \sqrt{250,000^2 + 963,375^2} = 1,245,000 \text{ inch-lbs.}$$

Diameter of journal

$$= \sqrt[3]{\frac{1,245,000}{8000}} \times 5{\cdot}1 = 9{\cdot}25 \text{ inches.}$$

Maximum equivalent twisting moment on aftermost journal

$$= 250,000 + \sqrt{250,000^2 + 1,333,500^2} = 1,606,700 \text{ inch-lbs.}$$

Diameter of journal

$$= \sqrt[3]{\frac{1,606,700}{8000}} \times 5{\cdot}1 = 10{\cdot}1 \text{ inches.}$$

The aftermost crank-arm will be 11 inches across the face; to find its thickness 18 inches from the pin.

Bending moment at that section = 66,675 × 18 = 1,000,000 inch-lbs.

$$\text{Thickness} = \frac{6 \times 1,200,150}{11^2 \times 8000} = 7{\cdot}44 \text{ inches.}$$

In actual practice the crank-shaft would not be made with the four journals all of different diameter, but some engineers make the shafts partly in accordance with theory, by arranging the two forward journals of the same diameter, and the two aftward journals of the same diameter; that is, for the example given above, the journals of the forward crank would be each 9·04 inches diameter, and those of the aftward one 10·1 inches diameter.

Example.—To find the dimensions of the crank-shaft of a single engine, whose cylinder is 30 inches diameter and stroke 50 inches, the steam used is 65 lbs. per square inch absolute pressure, and the cut-off at 0·3 the stroke. The distance between foundation facings for shaft brasses is 40 inches. The connecting-rod is 100 inches long. Back pressure and loss at piston are 5 lbs.

The maximum pressure on the piston is 60 × 706 = 42,360 lbs.

The maximum twisting moment occurs just at the cut-off in this case, and is 42,360 × 24, or 1,016,640 inch-lbs.

The bending moment on each journal at that period $\frac{42,360 \times 40}{8}$ or 211,800 inch-lbs.

The bending moment on the after arm, at a distance of 22 inches from the centre of crank-pin, is 42,360 × 22, or 931,920 inch-lbs.

Diameter of fore journal

$$= \sqrt[3]{\frac{211,800}{8000} \times 10 \cdot 2} = 6 \cdot 46 \text{ inches.}$$

Maximum equivalent twisting moment on after journal

$$= 211,800 + \sqrt{211,800^2 + 1,016,640^2} = 1,250,200 \text{ inch-lbs.}$$

Diameter of after journal

$$= \sqrt[3]{\frac{1,250,200}{8000} \times 5 \cdot 1} = 9 \cdot 27 \text{ inches.}$$

If the crank-arm is 10 inches wide at the face, then thickness of crank-arm at 22 inches from pin $= \frac{6 \times 931,920}{10^2 \times 8000} = 7$ inches.

The bending moment at the centre of the pin of a solid or rigidly built up crank-shaft is $\frac{R \times L}{8}$.

Note.—A solid shaft, or one whose continuity of strength is unbroken from end to end, is treated, so far as bending stresses are concerned, as a girder or beam *secured* at its points of support; or as a continuous girder supported at several points when there are more than two journals. Hence, the bending moment in the *middle* between two journals is $\frac{R \times L}{8}$; *and at the points of support* also $\frac{R \times L}{8}$, since change of flexure takes place at a distance $\frac{L}{4}$ from the supports. Marine crank-shafts whose arms are at least 0·7 of the diameter thick, and whose bearings *thoroughly* support the shaft close to the crank-arms, are really subject to little or no bending action *at the journals.*

At and near the end of the stroke the crank-arms are subject to a bending strain applied very suddenly when the steam enters the cylinder, on opening to lead after only slight *compression.* The force should be taken at twice the load on the piston (2 P), and if L_1 is the length of the pin + the thickness of the crank-arm as close to the shaft, then

$$\text{Bending mement} = \frac{2 P \times L_1}{4} = \frac{P \times L_1}{2}$$

Crank-shafts are subject to intermittent loads, and therefore to intermittent stresses (*v.* Table xlvi.).

And the bending moment on the section of *each* arm caused by this force is

$\frac{P \times L_1}{4}$ and *acts at right angles* to the bending force, due to the twisting force on the crank. Hence,

$$b = 6 \frac{(P \times L_1)}{4 (a^2 f)} = \frac{3 P \times L_1}{2 a^2 f}.$$

In the previous example L_1 may be supposed to be 23 inches, and P is 42,360 lbs., a is 7 lbs.; then

$$b = \frac{3 \times 42{,}360 \times 23}{2 \times 7^2 \times 8000} = 3{\cdot}6 \text{ inches.}$$

So that the forward crank-arm must not be less than this thickness at any part.

Crank-shafts for screw engines, when above 10 inches diameter, are generally made in duplicate pieces, so that in case of damage to one only a part of the shaft is condemned, and a spare piece can be easily carried on foreign voyages. And also by this plan there is less labour in replacing the damaged part, than if the whole shaft is moved.

Built Crank-shafts.—Shafts above 15 inches diameter are better built up than in one forging, and they can then be made of steel at an expense very little beyond that of iron; indeed, many engineers now make crank-shafts of all sizes on this principle. The crank-arms are of the same thickness at the pin as at the shaft, and equal to 0·7 to 0·8 of the diameter of shaft journals; the end view, as in fig. 58, shows the usual shape for large cranks—smaller ones are often straight on the sides. Great care is required properly to construct such a shaft so as to be perfectly true when finished, and to have the arms shrunk on sufficiently tight without leaving the metal around the pins, and shaft-ends in such a state of tension as to be dangerous.

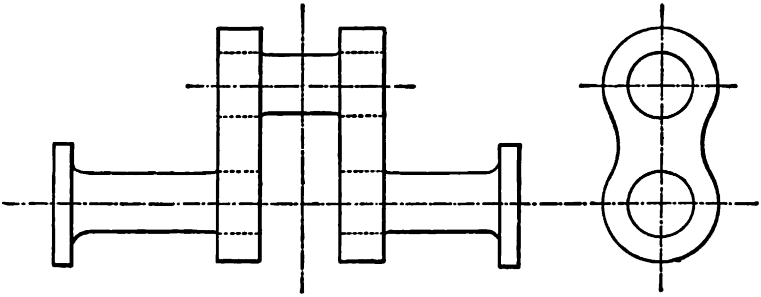

Fig. 58.—Built up Crank-shaft

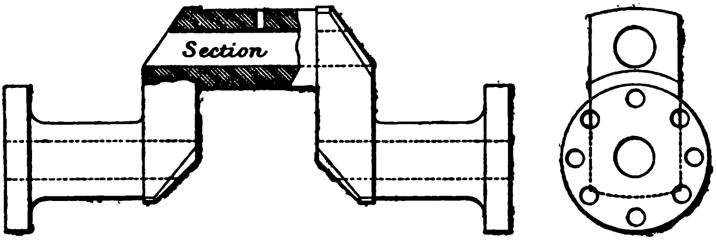

Fig. 59.—Naval Crank-shaft.

The thickness of the metal around the shaft, &c., can be calculated, as before stated for the overhung crank.

The crank-arms are sometimes forged with the shaft-ends, and the pins shrunk into eyes in the arms. This method has advantages, but it is very unsightly, and misses one of the chief merits of a built crank-shaft. Another arrangement is to make the crank-pin and arms in one piece, generally a steel casting, and shrunk it on the shaft-ends or shanks; and sometimes it is secured to the latter by flanges, bolts, &c.

There are a number of other patented forms of crank-shafts, some having the crank-arms of cast steel, and some of forged steel and iron, so arranged as to couple the shafts at the cranks instead of between them.

Fig. 59 shows a piece of crank-shaft as generally made for fast-running engines, and the dotted lines indicate the modifications made when further weight is to be saved, as in naval and other fast-running engines.

Couplings.—It is usual now to have the coupling forged with the shaft instead of keyed on as formerly. The tube shafts of twin-screw engines, however, generally have one coupling keyed on. As a rule, the only strain to which a coupling is subject is twisting; hence, if t be the thickness of the coupling, and r the distance of any part of the coupling from the centre of the shaft which is subject to a twisting moment T: the section of metal resisting the force is $2\pi r t$; and if f be the strain per square inch on this section, acting at the distance r, then

$$T = 2\pi r t f \times r = 2\pi r^2 . f . t, \text{ that is, thickness of coupling} = \frac{T}{2\pi r^2 f}.$$

If r is the radius of the shaft subject to twisting only, so that $\frac{\pi r^3}{2} f$ is equal to T. Then

$$\text{Thickness of coupling} = \frac{\pi r^3 f}{2} \div 2\pi r^2 f, \text{ or } \frac{r}{4}.$$

From practical considerations the thickness of the coupling should not be less than the diameter of the bolts, and since the strength of a coupling is somewhat impaired by the holes drilled through it for the bolts, it should not be less than 0·3 the diameter of the shaft subject to twisting only.

Coupling Bolts.—When shafts are close coupled, and the bolts are a good fit in the holes, they are subject to a shearing force only, caused by the twisting strain on the shaft; hence, if d be the diameter of the bolts, whose number is n, K is the distance from centre of bolts to centre of shaft, T the twisting moment, and D the diameter of the shaft subject to twisting only. Then

$$T = n\frac{\pi d^2}{4} f \times K; \text{ or } d = \sqrt{\frac{T}{0{\cdot}7854 f . K . n}};$$

but

$$T = \frac{\pi D^3}{16} f.$$

Hence,

$$\text{Diameter of bolts} = \frac{D}{2}\sqrt{\frac{D}{n \times K}}.$$

If K is always taken at 0·8 × D. Then

$$\text{Diameter of bolts} = \frac{D}{2}\sqrt{\frac{1}{0{\cdot}8 \times n}}.$$

Then when there are 5 bolts,

$$\text{Diameter of bolts} = \frac{D}{2}\sqrt{\frac{1}{5 \times 0{\cdot}8}} = \frac{D}{4}.$$

If 3 bolts, diameter of bolts = diameter of shaft ÷ 3·10.
" 4 " " = " 3·58.
" 5 " " = " 4·00.
" 6 " " = " 4·38.
" 7 " " = " 4·73.
" 8 " " = " 5·06.
" 9 " " = " 5·37.
" 10 " " = " 5·67.

The number of bolts in a coupling depends sometimes on circumstances, but usually there should be two, and an allowance of one more for each 2 inches of diameter of shaft, and the above proportions are based on this allowance; but when it is necessary to have the couplings as small as possible the number may be increased, and with the consequent decrease in diameter, the centres of bolts may be nearer to the centre of shaft.

The couplings of a two-crank engine, whose shaft is in duplicate halves at right angles, should have four or a multiple of four bolts; and for those of a three-crank engine, whose shaft is in three duplicate pieces, the number of bolts must be a multiple of three.

When the shafts are of mild steel, the bolts should be of a harder kind; indeed, as they are subject to shearing stresses only, they may be made of a steel 40 to 45 inches tensile strength.

Surface of Crank-pins and Shaft-journals.—Measuring, as in the case of gudgeons and crossheads, the effective bearing surface as the diameter multiplied by the length of the bearing, then the bearing surface of cranks was such that the pressure per square inch did not exceed 500 lbs.; the crank-pins of high-speed engines should, however, be such that the pressure per square inch does not exceed $\dfrac{6500}{\sqrt{R + 100}}$ lbs.; that is, if d is the diameter of the crank-pin, and l its length, L the maximum recurrent load on piston, R the revolutions per minute, then

$$d \times l = L \div \frac{6500}{\sqrt{R + 100}}; \text{ or } \frac{L\sqrt{R + 100}}{6500}.$$

When the brass is recessed, so that it bears only in parts on the shaft, the actual bearing surface should not be exposed to more than 600 lbs. pressure per square inch.

The pins of paddle-wheel engines, owing to the comparatively slow speed of shaft, may be designed to take a pressure of 800 to 900 lbs. per square inch.

The main bearings in which the crank-shaft runs should be such that the pressure never exceeds 600 lbs. per square inch, and for screw engines the pressure should not exceed 400 lbs.; when the brasses are recessed or fitted with white metal strips, the actual bearing or rubbing surface should be such that the pressure does not exceed 600 lbs. per square inch. The main bearings of screw engines, when room admits, should be such that the nominal pressure does not exceed $\dfrac{4300}{\sqrt{R + 100}}$, measuring the whole of the bearing, or 30 per cent. more, measuring only the parts on which the shaft bears. If, then, d be the diameter of the journals and l the length of each—

$$d \times l = \frac{L}{2} \times \frac{\sqrt{R + 100}}{4300}.$$

The length of the crank-pin is from 1 to 1·5 of the diameter, and that of each journal from 1 to 1·5 the diameter of the journal. Vertical engines have usually sufficient space for a crank-pin 1·25 the diameter, and each journal 1·5 the diameter. The foremost journal of a compound engine is often made much shorter than the others, to allow the eccentric sheaves to be nearer the centre, so as to come in line with the valve-rod.

Owing to the comparatively small pressure on the crank-pins and journals of three-crank compound, and triple- and quadruple-compound engines, the intermediate journals may be generally somewhat shorter; but the first and last journals should not be materially less than in a two-crank engine.

Drivers.—In order to avoid any of the thrust of the propeller coming on the crank-shaft and its bearings, the coupling-bolts connecting it to the thrust-shaft are sometimes made without heads, so that they are free to move in and out of the holes in the coupling flange of one of these shafts while held firmly in that of the other. When this is so they should be of larger diameter than the ordinary coupling bolts, and the part projecting from the face of the flange into which they are secured should be larger still, so as to form a shoulder, against which they may be tightened up, and to give the necessary strength to resist bending. The faces of the flanges when thus loosely coupled should be from ¼ to ½ inch apart. It was found necessary, generally, to provide means for lubricating these drivers, especially in heavily armoured and high-powered warships with single screws. Drivers are still necessary in all ships which are liable to change shape from varied loading, as in cargo steamers in heavy seas, and in long steamers of light build. As a rule, however, it is better to connect the crank and thrust-shafts in the ordinary way with bolts, &c., and to leave such clearance in the bearings and brasses that the crank-shaft may have some longitudinal "play."

Taper Bolts are often used in lieu of the ordinary parallel ones, especially for the shaft next the propeller-shaft, to facilitate their withdrawal. Taper bolts can be used with advantage when the flange is by necessity small, for the screwed end is much smaller than the part at the junction of the two shafts subject to shearing. These bolts also are necessarily a tighter fit in the hole, since the tightening of the nuts draws them farther into it.

Cross-keys are sometimes fitted to couplings. Half the key is bedded into a recess in the face of each flange, and so it takes the shearing stress from the bolts.

Since with a number of bolts or drivers it is possible, from wear or bad workmanship, that the strain is taken by only a part of them, it is usual to provide an excess of strength. This provision can be conveniently effected by proportioning them to the diameter of the *crank-shaft* as if it were subject to twisting only.

Propeller Shafts, sometimes called "screw" shafts, and sometimes "tail end" shafts. The propeller shaft is subject to the twisting stress of the engine, and to a bending stress due to the weight of the propeller. In rough weather, when the ship is pitching, the strains are increased and become very severe; for when the screw is partially immersed, in addition to the twisting moment by the reaction of the water acting on one side only, there is a bending stress as on a paddle-shaft; besides which the momentum of the screw when pitching also adds severe bending stresses, all of which are moreover alternating.

In still water the bending moment on the shaft is the weight of the screw multiplied by the distance of its centre from the stern bush. To provide for the strains in rough weather, the bending moment should be taken at twice this value; for ships which may cross the Atlantic at any time in ballast trim, even this is insufficient to produce a shaft sufficiently large to last a satisfactory length of time. Moreover, the material of a shaft working under such conditions is subject to alternating stresses of considerable intensity, which tend to degrade it and render it unfit to resist such strains as come on the shaft. The diameter of the screw is also an important factor in determining the size of screw shafts of ships subject to rough weather.

Hence, if T is the maximum twisting moment on the crank-shaft, W the weight of the propeller in pounds, and L the distance of its centre from the stern bush,

$$\text{Maximum bending moment} = 2\,W \times L;$$

and

$$\text{Maximum equivalent twisting moment } T^1$$
$$= 2\,W \times L + \sqrt{(2\,W \times L)^2 + T^2};$$

and as before,

$$\text{Diameter of screw-shaft} = \sqrt[3]{\frac{T_1}{f}} \times 5{\cdot}1.$$

Example.—To find the diameter of the screw-shaft for an engine whose maximum twisting moment is 1,333,500 inch-lbs. The weight of the screw is 6000 lbs., and the distance of its centre from stern bush is 20 inches.

The max. bending moment

$$= 2 \times 6000 \times 20 = 240{,}000 \text{ inch-lbs.}$$

The max. equivalent twisting moment

$$= 240{,}000 + \sqrt{240{,}000^2 + 1{,}333{,}500^2} = 1{,}594{,}000 \text{ inch-lbs.}$$

Diameter of shaft

$$= \sqrt[3]{\frac{1{,}594{,}000}{6000}} \times 5{\cdot}1 = 11{\cdot}2 \text{ inches.}$$

It is such a very serious matter when the screw-shaft breaks, that it should always be of ample size, and for ships in the Atlantic trade it should be specially strong. It was usual to make it the same diameter as the crank-shaft, but in some ships even this is not sufficient, and it is now not at all an unusual thing to make them 20 per cent. stronger than the crank-shaft.

When the screw is fitted in a "banjo" frame for lifting above water when the ship is under sail, the shaft is, of course, nearly wholly free from bending stresses.

Outer Bearing.—It was customary to provide an outer bearing on or in the rudder-post, for the extreme end of the screw-shaft to rest upon; but since the rudder-post gives no support sideways, and a very precarious one in any direction, the practice is now obsolete. Also, it has been found that when ships so fitted have touched the ground with the heel, the screw-shaft was bent, and sometimes dangerously so. The strongest argument in favour of this outer bearing is, that it prevents the loss of the screw when the shaft is broken; but if the shaft is broken, and the ship has to depend on the sails, it is better to be without the screw; and if the shaft is broken diagonally, and the screw is caused to revolve from the motion of the ship, there is great risk of splitting the stern tube. If there is no outer bearing, and the fracture is well within the tube, the screw will not be lost, but will go back until the shaft-end butts against the rudder-post, and revolves then without danger. If the shaft breaks close to the propeller, and there is an outer bearing, the danger of damage to the rudder-post is very great indeed, from the wrenching of the bearing on the propeller falling out.

Screw-shaft End.—The shaft end fitted into the screw-boss should be turned to a taper of ¾ inch to the foot; if the taper is less than this, as is sometimes the case, extreme difficulty is experienced in getting the screw off. The screw should be secured by a key extending the whole length of the boss, and driven into place after the screw is thoroughly well driven on. The screw is retained in place by a nut, the screw-thread of which is the reverse of that of the screw itself. A tail key through the shaft end was preferred by some engineers as a means of retaining the screw in place; but although it is a very safe plan, it is not so convenient as the nut. When a nut is employed a safety key or pin is fitted in rear of it, or else a set-screw or other simple means of locking it is used. These nuts are now often of cast steel or bronze, and made so as to cover the end of the shaft. When the propeller is of bronze a "cap" nut of the same material is necessary. Such nuts are secured by set-screws in the end "out of centre," or set-screws in the boss-fitting in recesses in the side of the nut.

Screw-shafts are encased with brass from the propeller to the inner end of the stern tube in H.M. Navy; in merchant ships this was only occasionally the case, partly on account of the expense, and partly because it prevents examination of the shaft and the detection of flaws, which may extend unobserved until rupture takes place. Brass casings, on the bearing parts, are used not so much to protect the shaft from corrosion, as on account of its wearing better when running on lignum vitæ, and admitting of wear without weakening the shaft thereby. Lloyds now require casings the full length.

When working in sandy water lignum vitæ wears very quickly and grinds away the brass casing. Ships which are often exposed to this are better without the brass casing and the lignum vitæ, and should be fitted in lieu with Fenton's white metal or cast-iron bush, and the shaft either without casing or cased with an iron liner, which may be renewed when worn.

The Stern Bush should be of such length that the pressure per square inch (measured as stated for bearings) does not exceed 35 lbs., and generally without inconvenience it may be arranged that the pressure is only 25 to 30 lbs. The pressure is, of course, in this case that due to the weight of the propeller, together with half that of the screw-shaft. The stern bush in practice is of a length equal to two to four times the diameter of bore. The stern-shaft should be supported on a bearing in the tunnel when possible; when this is either not possible or inconvenient, it may rest on a bush in the stern tube just beyond the stuffing-box.

In the mercantile marine the screw-shaft, when partly cased with brass, has bushes fitted with lignum vitæ; the brass casings extend from the screw boss to an inch or two beyond the inner end of bush, and also where the shaft passes through the stuffing-box, and inner bush when there is one. Of late years there has been a tendency to follow the Admiralty method of casing in the whole length covered by the stern tube. Certain of the Register Societies have favoured this course, in spite of the prevention of inspection thereby. In order to make the process as easy and inexpensive as possible, the stern tube is made very short, so that the brass casing is very little longer than the sum of the lengths of the two liners as formerly fitted.

The Thickness of Brass Casing at bearing parts is 0·3 inch + 0·035 × diameter. In order to easily withdraw the shaft, the diameter of the inner casing should be $\frac{1}{8}$ inch larger than the outer.

Stern Tube.—In the Navy the stern tube is always of brass, and in iron, steel, and composite ships is within another wrought-iron or steel tube secured to the framing of the ship. The lignum vitæ strips are fitted into the tube, either in separate grooves for each strip, or the strips fit in side by side with a brass strip at the top secured by screws to the tube which keys them so as to form a bush. A similar but shorter set of strips are fitted next the stuffing-box. The brass stern tube fits into the stern-post accurately and tightly, and is secured to the bulkhead by a flange, &c.*

In the mercantile marine the stern tube is nearly always of cast iron, whose thickness is 0·5 inch + 0·08 the diameter. The stern bush of brass fits accurately into the tube, and the stuffing-box neck ring and gland are lined with brass. There are various ways of fitting the stern tube in place.

The common plan adopted by most engineers is to turn the outer end so as to fit accurately and tightly into the hole bored in the stern-post, and secure it by a nut screwed on its end. The inner end has a flange, and next it a projecting rim, which is turned to fit in the hole bored in the bulkhead; the flange is bolted to the bulkhead after a liner is fitted between them.

The tube is sometimes bolted to the stern-post by two lugs cast with it, one above and one below it.

Another plan of fixing the stern tube is to fit its outer end into a recess bored in the stern-post, and secure it by bolts to the bulkhead as before described, and by two strong draw-bolts passing through the flange to a partial bulkhead two or three frame spaces nearer the stern. In this case the stern bush is partly in the tube and partly in the stern-post.

Stern Bushes.—When made of white metal they should be of a thickness = 0·5 inch + 0·03 × diameter. Those fitted with lignum vitæ are of brass, and formed with a flange, which is secured to the stern tube by screws, which prevent it from turning or coming out. Stern bushes are often made of strong cast-iron, with white metal strips driven in, or with it run in as in a main bearing. Such bushes have flanges like the bronze ones. The lignum vitæ is sometimes fitted in strips, as in brass stern tubes, and sometimes into square holes, the bush being cast as a skeleton to hold the wooden blocks. This latter plan is very convenient for small ships, but not a good one for large ones, as the wood by the continued concussion gets impressed on the cast-iron tube. Lignum vitæ wears best in end grain, especially when it is of inferior quality; when cut from a good tree of large size it wears equally well either way.

Lignum vitæ and white metal strips should be from $\frac{3}{8}$-inch to $\frac{3}{4}$-inch thick, and about three to four times their thicknesses in breadth; they must be bevelled so as to leave free watercourses between them. The brass behind the strips should be 0·04 × diameter in thickness, and the metal ridges between the wooden strips of the same thickness. Sometimes the brass bush

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* It is usual, however, to fit a separate bush containing the lignum vitæ strips in all but small ships for convenience of repair.

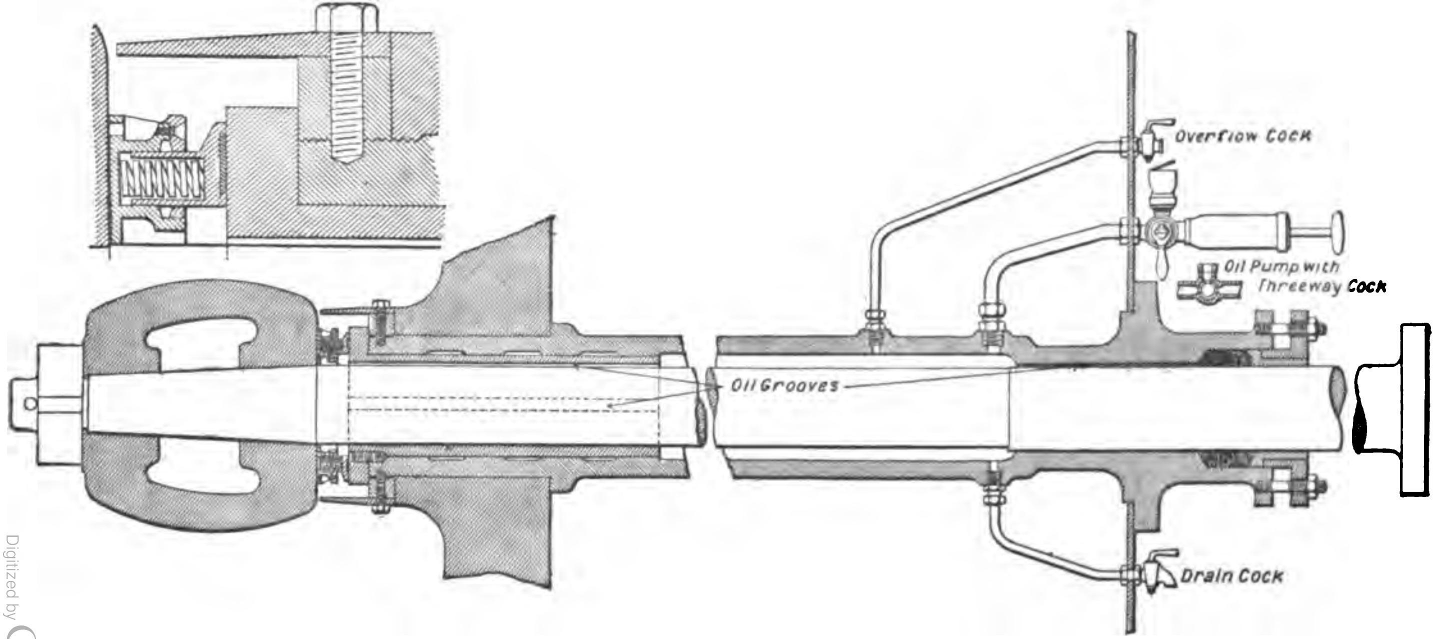

Fig. 59a.—Screw-shaft and Stern Tube. Cedervall's Patent for Oil Lubrication.

is dispensed with, and the strips fitted into the cast-iron tube as in the brass tube, but this is not good practice.

A pipe is fitted leading from the top of the stern tube to the bulkhead, through which the water may run from the tube so as to cause a fresh supply to enter from the sea, and thereby prevent heating.

From time to time attempts have been made to exclude sea water from the stern tube, and lubricate the shaft bearings with oil or soapy water. Ordinary stuffing-box, gland, &c., have been fitted to the tube end, and the lubricant fed through a pipe from the upper deck, so as to give a "head" superior to that of the sea water, and so prevent any leakage inward. No great degree of success has followed this plan, as no provision is made for the wear of the bush. But if the bush did not wear the violent action of the screw in a seaway would inevitably cause leakage. This difficulty has been got over by Mr. Cedervall, who prevents water from entering by the means shown in fig. 59*a*, which has been generally successful in practice.

Thrust-shaft.—Although the crank-shaft was sometimes made with a collar or collars on it, to take the thrust, it is not good practice, especially for large engines. The crank-shaft should be required only to take the strains from the pistons, and be free to move around in its bearings without end pressure; and since any longitudinal displacement of the crank-shaft tends to throw abnormal strains on the working parts, it is better to remove all causes of such a derangement. To this end the thrust collars should be on one of the intermediate shafts, and for convenience on that one next the crank-shaft. If possible the thrust bearing should be in the engine-room, and it is for this purpose chiefly that the collars were sometimes on the crank-shaft.

Thrust.—To find the thrust along the shaft of a screw engine, it is necessary to know the speed of the ship and the effective horse-power. The effective horse-power is the power actually employed in propelling the ship, and of course its relation to the *indicated* horse-power depends on the combined efficiency of the engines and propeller. This ratio varies from 0·77 with the best high-speed engines to 0·68 with the ordinary merchant steamers' machinery. With turbines it is probably 0·8. If P be the pressure in pounds exerted by the propeller against the thrust bearing, and S the speed of the ship in feet per minute, then

Work done in moving the ship $= P \times S$,

and therefore $$\text{Effective H.P.} = P \times S \div 33{,}000,$$

or $$\text{I.H.P.} \times f = \frac{P \times S}{33{,}000},$$

then $$P = \text{I.H.P.} \times \frac{33{,}000}{S} \times f.$$

Now, if K be the speed in knots,

$$S = K \times \frac{6080}{60}.$$

Then

$$P = \text{I.H.P.} \times \frac{326 \times f}{K}.$$

P is called the *mean normal thrust*.

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Example.—To find the thrust on the shafting of an engine whose I.H.P. is 2000, and the speed of the ship 12 knots, the efficiency 0·66.

$$P = 2000 \times \frac{326 \times 0{\cdot}66}{12} = 36{,}166 \text{ lbs.}$$

Now, it will be seen that P varies with the I.H.P., and inversely as the speed, so that the thrust of a particular screw may vary very considerably; for if from some cause the speed is decreased, without a corresponding decrease in the power, the thrust must of necessity increase. This actually occurs in practice, and must be provided for always. The times when the actual thrust exceeds the normal thrust are when the engine first moves, and its power is employed in overcoming the inertia of the ship, also when the ship is towing, and when driving against a head wind or sea. It is also to be noted that the speed of ship means speed *through the water;* for it is on this account that so little strain comes on the moorings of a ship whose engines are working at full speed when in a dock or confined piece of water. In this case it is only at first starting that any great strain is thrown on the moorings, for as soon as the water is set into motion so as to flow past the ship in a steady stream, the power is absorbed in facing the stream and really propelling the ship through the water.

Another cause of variation in the thrust is the variation in the twisting moment, which is, as before shown, very great in certain classes of engines.

The surface exposed to thrust may, however, be calculated from the mean normal thrust, and allowance made for all emergencies. This surface should be such that the pressure per square inch from the mean normal thrust does not exceed 70 lbs.; and for tugboats or ships especially exposed to severe weather, or service analogous to either of these, it should not exceed 50 lbs. Ordinary merchant ships have usually such surface that the nominal thrust does not exceed 60 lbs., and naval ships 45 lbs., as they may have to tow or do analogous work at full speed.

A good rule is as follows:—Pressure per square inch of thrust $= 800 \div \sqrt{R\,d + 100}$, R being the revolutions and d the diameter of thrust-shaft in feet.

Diameter of Thrust Collars.—Let P be the mean normal thrust, d the diameter of the shaft, and D the diameter of the thrust collars, whose number is n, and the pressure 60 lbs. Then

$$P = 60\left(\frac{\pi D^2}{4} - \frac{\pi d^2}{4}\right) n = 47\,(D^2 - d^2)\,n;$$

and

$$D = \sqrt{d^2 + \frac{P}{47\,n}}.$$

The thickness of each collar for mere strength

$$= \frac{P}{n} \div (\pi d \times 1000) = \frac{P}{3142\,d\,n}.$$

In practice the thickness of each collar $= 0{\cdot}4\,(D - d)$.

(1) Space between the collars, if rings are of solid brass $= 0{\cdot}4\,(D - d)$.

(2) Space between the collars, if rings are of cast iron faced with brass or white metal $= 0{\cdot}75\,(D - d)$.

(3) Space between the collars, if rings are of hollow brass for water to circulate through $= D - d$.

The number of collars depends very much on the size of the engine and the prejudice of the designer. If there are many collars, they are of necessity somewhat small, and although the *chances* are in favour of the majority of them acting efficiently, allowance must be made for the contingency of the whole thrust coming on only one of them, and the larger the number of collars, the less able is each one separately to resist the whole thrust. The chief objection to a few collars is, that they are of necessity of comparatively large diameter, and have, therefore, a greater resistance to turning owing to the longer radius, besides a higher speed of rubbing surface; there is also the consideration of cost of forging against large collars.

When there are a few large collars a better design of thrust-block is possible, and the rings can be made adjustable without removal.

The number of collars should vary with the size of the shaft, and a very good rule is, that there should be one collar for shafts up to 6 inches diameter, and then an additional collar for every 2 inches of diameter beyond this when the collars are large, and the engines slow running; fast-running naval engines require more than this to get sufficient surface.

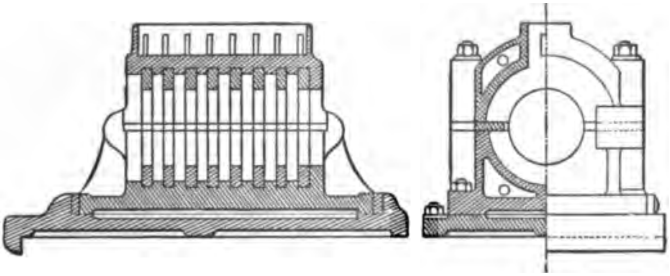


Fig. 60.—Common Thrust-block.

The common plan of thrust-block with many collars is shown in fig. 60, and a modification of the same is made by having the rings in one casting. These plans are cheap, and do very well for small engines. So long as no heating is allowed to take place in the bearing, it will work very well; but when once it gets out of order it is difficult to deal with and impossible to adjust at sea. It suffers from being enclosed, and from the rings lacking means of independent adjustment. Fig. 61 shows a plan of thrust-block which is most suitable when there are a few large collars. Here the thrust is taken by horse-shoe shaped pieces of metal faced with bronze or white metal, and fitted sometimes carefully into recesses on either side of the main block. When faced with bronze, each may be adjusted very simply by putting thin tin liners behind the facings, which are hung on steady pins. Phosphor bronze or gun metal does fairly well for thrusts when the shafts are of wrought iron, but white metal does better and must be employed when the shafts are of mild steel.

Fig. 61 is an elaboration of the form of block. Here the horse-shoes fit over two screwed bars, one on either side of the block; nuts are fitted to these bars, so that each collar may be adjusted by its own nuts, or the whole of them by the nuts at the end. A simpler and cheaper method is to turn these side bars so that there is a collar between each pair of shoes instead of a pair of nuts, that is, the bars have a set of collars corresponding to those of the shaft, and take the thrust of the shoes as before.

Both these plans are most successful in practice, in great measure due to the fact that the collars are open and exposed at the top, so as to be easily lubricated and cooled by the air, and to their running in oil, or in a mixture of oil and soapy water contained in the trough below them.

It is most important that a bearing be placed close to the thrust, so that the shaft cannot vibrate and cause uneven pressure over the surface of the collars. The function of the thrust bearing is to take only end pressure. This is particularly the case when designed with horse-shoe rings.

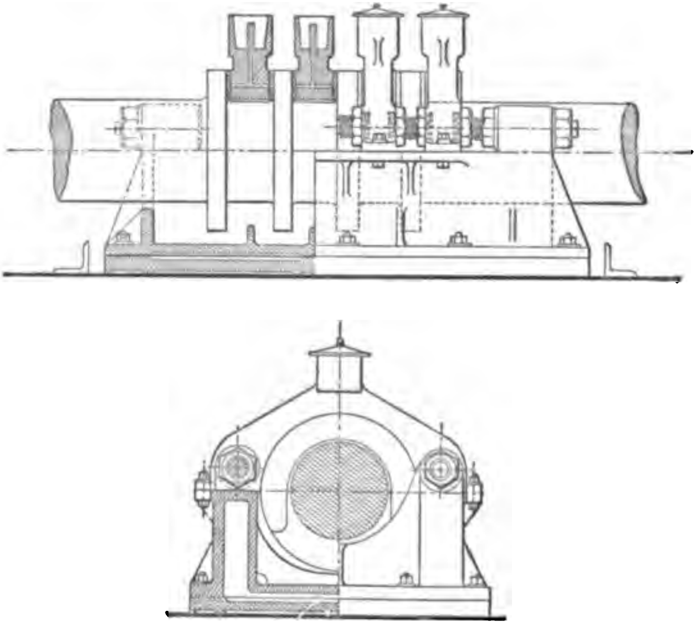


Fig. 61.—Improved Thrust-block.

Diameter of Shafts, Rules for, in Practice.—As almost every ship is classed with one or other of the shipping registers, the shafting must in no case be smaller than provided for in the rules of the corporation with which she is classed, and from whom a machinery certificate is necessary. All British ships carrying more than 12 passengers must have a certificate from the Board of Trade, whose surveyor has to report that, *inter alia*, the shafting is fit and proper, &c.; in his case it means that the diameters are not less than given by the Board's Rules.

The Admiralty now generally specify the sizes of the various shafts, which are almost invariably hollow, and of steel.

BOARD OF TRADE RULES FOR SHAFTS.

For compound condensing engines with two or more cylinders, when the cranks are not overhung:—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{f\left(2 + \frac{D^2}{d^2}\right)}}$$

$$P = \frac{f \times S^3}{C \times D^2}\left(2 + \frac{D^2}{d^2}\right)$$

Where S = diameter of shaft in inches.
d^2 = square of diameter of high-pressure cylinder in inches or sum of squares of diameters when there are two or more high-pressure cylinders.
D^2 = square of diameter of low-pressure cylinder in inches or sum of squares of diameters when there are two or more low-pressure cylinders.
P = absolute pressure in lbs. per square inch—that is, boiler pressure plus 15 lbs.
C = length of crank in inches.
f = constant from following table.

Note—Intermediate pressure cylinders do not appear in the formulæ.

For ordinary condensing engines with one, two, or more cylinders, when the cranks are not overhung:—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{3 \times f}},$$

$$P = \frac{3 \times f \times S^3}{C \times D^2}.$$

Where D^2 = square of diameter of cylinder in inches; or sum of squares of diameters where there are two or more cylinders. Other symbols as above.

TABLE XVII.

For Two Cranks—Angle between Cranks.	For Crank and Thrust Shafts. f		For Tunnel Shaft. f	For Propeller Shaft.* f
90°	For paddle engines of ordinary type multiply constant in this column suitable for angle of cranks by 1·4.	1047	1221	890
100°		966	1128	821
110°		904	1055	768
120°		855	997	727
130°		817	953	694
140°		788	919	670
150°		766	894	651
160°		751	877	638
170°		743	867	631
180°		740	864	629
For Three Cranks.				
120°		1110	1295	943

Note.—When there is only one crank the constants applicable are those in the Table opposite 180°.

* The portion of the propeller shaft which is forward of the stern gland, and all the thrust shaft, with the exception of the part enclosed in the thrust bearing, may be the same diameter as the intermediate tunnel shafting.

LLOYD'S RULES WITH REGARD TO SHAFTING.

The diameters of crank and straight shafts are to be not less than those given by the following formulæ :—

TABLE XVII*a*.—DIAMETERS OF SHAFTS.

Description of Engine.	Diameter of Intermediate Shaft in Inches.
Compound — Two cranks at right angles, - -	$(\cdot 04\ A + \cdot 006\ D + \cdot 02\ S) \times \sqrt[3]{P}.$
Triple—Three cranks at equal angles, - -	$(\cdot 038\ A + \cdot 009\ B + \cdot 002\ D + \cdot 0165\ S) \times \sqrt[3]{P}.$
Quadruple — Two cranks at right angles, - -	$(\cdot 034\ A + \cdot 011\ B + \cdot 004\ C + \cdot 0014\ D + \cdot 016\ S) \times \sqrt[3]{P}.$
Quadruple—Three cranks,	$(\cdot 028\ A + \cdot 014\ B + \cdot 006\ C + \cdot 0017\ D + \cdot 015\ S) \times \sqrt[3]{P}.$
Do. —Four cranks,	$(\cdot 033\ A + \cdot 01\ B + \cdot 004\ C + \cdot 0013\ D + \cdot 0155\ S) \times \sqrt[3]{P}.$

Where A is diameter of H.P. cylinder in inches.
B „ first I.P. „
C „ second I.P. „
D „ L.P. „
S is stroke of pistons in inches.
P is boiler pressure above atmosphere in lbs. per square inch.

The diameter of the crank-shaft and the *thrust-shaft* between the collars to be at least $\frac{21}{20}$ of that of the intermediate shafts. The diameter of thrust-shaft may be tapered off at each end to the same size as that of the intermediate shafts.

The diameter of the screw-shaft to be equal to the diameter of intermediate shaft multiplied by $\left(\cdot 63 + \frac{\cdot 03\ P}{T}\right)$, where P is the diameter of the propeller and T that of the intermediate shaft, both in inches. In no case, however, must the diameter of the screw-shaft be less than 1·07 T.

Note.—This size of screw-shaft is intended to apply when continuous brass liners are fitted the whole length of stern tube. If no liners are used, or if two separate ones, then the shaft must be $\frac{41}{40}$ of that given by the Rule. Further, these Rules apply only to two-cylinder compound engines when the ratio of low pressure to high pressure does not exceed 4·5 ; to triple-expansion engines when it does not exceed 9·0 ; to quadruple engines when it does not exceed 12·0 ; and when in all cases the stroke is neither less than half nor greater than the diameter of low-pressure cylinder.

THE BRITISH CORPORATION RULES.

Diameter of Shafting.—The minimum diameters of crank, thrust, propeller, and intermediate shafts may be found from the following formulæ, except where the ratio of length of stroke to distance between main bearings is unusual, when they will receive special consideration :—

$$D = \sqrt[3]{\frac{P \times L^2 \times S}{C}} \times B.$$

Where D = diameter of shaft.
P = absolute pressure—*i.e.*, boiler pressure + 15 lbs
S = stroke of engine, in inches.
L = diameter of low-pressure cylinder, in inches
B = 1·0 for crank- and thrust-shafts.
B = 0·95 for intermediate shafts.
B = for propeller shafts to be taken from the following Table :—

TABLE XVII*b*.

Coefficient of Displacement of Vessel at four-fifths Moulded Depth.	Ratio of Diameter of Propeller to Diameter of Crank-shaft.					
	13	14	15	16	17	18
·6	1·0	1·01	1·02	1·03	1·04	1·05
·62	1·01	1·02	1·03	1·04	1·05	1·06
·64	1·02	1·03	1·04	1·05	1·06	1·07
·66	1·03	1·04	1·05	1·06	1·07	1·08
·68	1·04	1·05	1·06	1·07	1·08	1·09
·70	1·05	1·06	1·07	1·08	1·09	1·10
·72	1·06	1·07	1·08	1·09	1·10	1·11
·74	1·07	1·08	1·09	1·10	1·11	1·12
·76	1·08	1·09	1·10	1·11	1·12	1·13
·78	1·09	1·10	1·11	1·12	1·13	1·14
·80	1·10	1·11	1·12	1·13	1·14	1·15

The value of the divisor C in the formula depends on the ratio $\frac{L^2}{H^2}$, where L = diameter of low-pressure cylinder and H of high-pressure cylinder, in inches :—

$\frac{L^2}{H^2}$	Two Cranks at 90°, Compound or Quadruple, also Three Cranks at 120°, Quadruple Expansion.	Three Cranks at 120°, Triple Expansion.	Four Cranks at 90°, Quadruple Expansion.
Ratio 3	9,910	...	...
,, 3⅛	10,160	...	...
,, 3¼	10,410	...	...
,, 3⅜	10,660	...	...
,, 3½	10,910	...	...
,, 3⅝	11,160	...	...
,, 3¾	11,410	...	...
,, 3⅞	11,660	...	...
,, 4	11,910	...	...
,, 4⅛	12,160	...	...
,, 4¼	12,410	...	...
,, 4⅜	12,660	...	...
,, 4½	12,910	13,650	...
,, 4¾	13,375	14,160	...
,, 5	13,840	14,670	...
,, 5¼	14,305	15,180	...
,, 5½	14,770	15,690	...
,, 5¾	15,235	16,200	...
,, 6	15,700	16,710	...
,, 6½	16,630	17,730	...
,, 7	17,560	18,630	...
,, 7½	18,410	19,530	...
,, 8	19,260	20,430	22,660
,, 8½	20,110	21,330	23,660
,, 9	20,960	22,200	24,660
,, 9½	21,750	23,070	25,660
,, 10	22,540	23,940	26,580
,, 10½	23,330	24,810	27,500
,, 11	24,120	25,660	28,420
,, 11½	24,900	26,500	29,340
,, 12	25,680	27,340	30,260

Rules of the Bureau Veritas.

Iron Shafts for Screw Steamers.

(*a*) *Crank Shafts.*—§ 7. When the crank of a screw engine is not overhung, the diameter of the shaft shall be determined by one of the following formulæ:—

For non-compound condensing engines:

$$d = \sqrt[3]{\frac{n \, \mathrm{P} \, \mathrm{L} \, \mathrm{D}^2}{\mathrm{C}}} \quad . \quad . \quad . \quad . \quad . \qquad \text{(A)}$$

For double, triple, and quadruple-expansion engines:

$$d = \sqrt[3]{\frac{\mathrm{P} \, \mathrm{L} \, (n_1 \, \mathrm{D}_1^2 + 0{\cdot}1 \, n \, \mathrm{D}^2)}{\mathrm{C}}} \quad . \qquad \text{(B)}$$

For shafts having a single overhung crank, the form under the radical sign is to be multiplied by

$$s + \sqrt{s^2 + 1}.$$

For two-cylinder single-crank tandem engines the formula will therefore be:

$$d = \sqrt[3]{\frac{\mathrm{P} \, \mathrm{L} \, (\mathrm{D}_1^2 + 0{\cdot}1 \, \mathrm{D}^2)(s + \sqrt{s^2 + 1})}{\mathrm{C}}} \qquad \text{(C)}$$

In those formulæ:

d = diameter of the after shaft bearing in inches.

n_1 = number of high-pressure cylinders.

D_1 = diameter of each high-pressure cylinder in inches. If there are several high-pressure cylinders the diameters of which are not the same, $n_1 \, \mathrm{D}_1^2$ represents the sum of the squares of their respective diameters.

n = number of low-pressure cylinders.

D = diameter of each low-pressure cylinder in inches. If there are several low-pressure cylinders the diameters of which are not the same, $n \, \mathrm{D}^2$ represents the sum of the squares of their respective diameters.

N.B.—For triple or quadruple-expansion engines the intermediate cylinders do not come into account in the formulæ.

L = length of stroke in inches, common to all pistons.

P = boiler pressure above atmosphere in pounds per square inch.

$s = \frac{a}{r}$ (see below). In order to determine a, B is supposed to be situated half-way the length of the bearing, unless the latter be longer than 1½ times the diameter; in this case B C may be considered as being equal to ¾ of the diameter.

C = a constant, the values of which for iron shafts are given below for certain cases. The values given apply to navigation in a seaway; for smooth water (tugs excepted) the constants may be increased by 30 per cent.

If it is above 15″, it should be increased by an amount to be determined by the Administration; for built-up shafts, however, this latter increase will not be required.

For hollow shafts the diameter must be increased by

1 per cent. if the diameter of the hole is 0·4 of the outside diameter.
2 „ „ „ 0·5 „
5 „ „ „ 0·6 „
10 „ „ „ 0·7 „

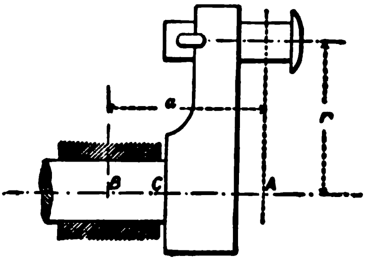


Fig. 62a.

If the hole is under 0·4 of the outside diameter, no increase will be required.

The Administration may allow a reduction on the diameter in certain special cases, for instance in well-balanced engines with light moving parts, or for very superior workmanship, &c. On the other hand, the Administration may require an augmentation for engines which differ much from the average proportions found in practice, thus, for instance, for engines having a comparatively small stroke; for compound engines, the low-pressure cylinder of which has a very large size compared with the high-pressure cylinder, &c.

VALUES OF C IN FORMULÆ (A), (B), (C).

I. *Non-Compound Condensing Engines with two Cylinders and two Cranks.*

Formula (A) where $n = 1$ and $C = 4800$ when the angle between the cranks is 90°. When it is not 90°, the constants are to be multiplied by the following coefficients :—

For 100°	0·91.
120°	0·79.
140°	0·72.
160°-180°	0·70.

II. *Double-Expansion Engines (Ordinary Compound).*

1. Two-crank two-cylinder intermediate receiver engines.

Formula (B) where $n_1 = n = 1$
$C = 3400$ when the angle between the cranks is 90°.

For other angles the constant is to be multiplied by the coefficients given in Case I.

2. Two-crank double-tandem engines (two high- and two low-pressure cylinders).

Formula (B) where $n_1 = n = 2$
$C = 3500$ when the angle between the cranks is 90°.

For other angles the constant is to be multiplied by the coefficients given in Case I..

3. Three-crank triple-tandem engines (three high- and three low-pressure cylinders).

Formula (B) where $n_1 = n = 3$
$C = 3800$ when the cranks are at 120°.

4. Single-crank tandem engines (one high- and one low-pressure cylinder). When the crank is *not* overhung:

Formula (B) where $n_1 = n = 1$
$C = 2100$.

When the crank *is* overhung:

Formula (C) where
$C = 3200$.

These two values suppose a cut-off at 0·8 of the stroke in the high-pressure cylinder. For later cut-offs the constant is to be increased.

5. Three-crank three-cylinder engines (one high- and two low-pressure cylinders).

Formula (B) where $n_1 = 1$, $n = 2$
$C = 3600$ when the cranks are at 120°.

III. *Triple-Expansion Engines.*

1. Three-crank three-cylinder engines (one high-, one intermediate, and one low-pressure cylinder).

Formula (B), where $n_1 = 1$, $n = 1$
$C = 3900$ when the cranks are at 120°.

2. Two-crank three-cylinder engines (two of the cylinders placed tandemwise).

Formula (B), where $n_1 = n = 1$
$C = 3000$ when the cranks are at right angles.

3. Two-crank four-cylinder engines (two tandem engines, viz., two high-pressure cylinders on the top of the intermediate and low-pressure cylinders).

Formula (B), where $n_1 = 2$, $n = 1$
$C = 3300$ when the cranks are at right angles.

IV. *Other Cases.*

For quadruple-expansion engines, as well as for other cases not included in the foregoing, the constant will be determined by the Administration.

N.B.—For quadruple-expansion double-tandem engines with two cranks at 90°, the value of C may be provisionally taken as 3100.

In the case of four cranks at right angles, the value of C may be taken at 3600. If the sequence of cylinders and the angles of the cranks are chosen so as to reduce the maximum torsional moment, the above value may be increased, but in no case shall exceed 4100. Each particular case to be submitted for approval.

(*b*) *Propeller, Tunnel, and Thrust Shafts.*—The diameter of propeller shaft must be $(1{\cdot}7\,\frac{D}{d} - 15)$ per cent. in excess of the diameter of the crank-shaft calculated from one of the formulæ (A), (B), or (C); D being the diameter of the propeller in inches, and d the diameter of the crank-shaft in inches. It is recommended to fit the propeller shaft in such a way that it cannot move endways, if for some reason or other it has been uncoupled from the rest of the shafting. Liners fitted on propeller shafts to be tapered off at ends.

For tunnel shafts a reduction of 6 per cent. on the diameter of the crank-shaft will be allowed.

The diameter of thrust shaft at the bottom of the collars, both between and immediately beyond these latter, to be equal to that of the crank-shaft, and tapered off at each end to the smaller diameter of the body of the shaft. The thrust of the screw propeller must be taken up by an efficient thrust block, so as to prevent any fore and aft strain on the crank-shaft.

Iron Shafts for Paddle Steamers.

§ 8. In side wheel steamers having double, triple, or quadruple-expansion engines with an intermediate shaft, each end of which carries an overhung crank-pin fitting loosely into an eye of the paddle shaft crank, the bearing of the latter (see A of the following sketch) must have its diameter calculated from the formula :

$$d = \sqrt[3]{\frac{P\,L\,(n_1\,D_1^2 + 0{\cdot}1\,n\,D^2)\,(s + \sqrt{s^2 + 1})}{C}} \quad . \qquad (D)$$

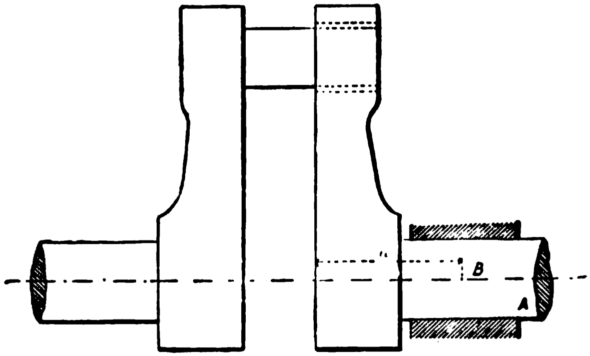


Fig. 62*b*.

where the letters have the same meaning as before (§ 7), except that a, for determining $s = \frac{a}{r}$, is to be measured as shown in the above sketch, the point B being the middle of the bearing.

For two-cylinder compound receiver engines with two cranks at 90°.

C = 13,000 for navigation in smooth water.

For coasting or sea-going steamers the constant is to be reduced, but need not be under 5700.

The diameter of the outer bearing of the paddle shaft and of the intermediate shaft to be submitted to the Administration or the Surveyors for approval. The same applies to other cases not dealt with in this paragraph.

Steel Shafts.

§ 9. Shafts made of steel must have a tensile strength of 26 to 30½ tons, and test pieces cut from the forging must satisfactorily withstand the prescribed tests. When it is desired to make use of steel of softer or harder quality, the corresponding alteration in the diameter must be submitted to the Administration (of the Bureau Veritas) for approval.

Summary.—The following is a summary of certain parts of this Chapter which will be useful for reference:—

$$\text{Rule 1.—Diameter of shaft} = \sqrt[3]{\frac{\text{I.H.P.}}{\text{revolutions}} \times \text{F}}.$$

The value of F is got from the following Table, where p is the boiler pressure—that is, the load on the safety valve:—

TABLE XVIII.—SHAFTS FOR SCREW ENGINES.

Description of Screw Engines.	Value of F Crank-shafts.	Value of F Tunnel-shafts.	Value of F Screw-shafts.
Single-crank single-cylinder, cut-off 0·5	150	130	175
,, ,, ,, ,, 0·2	200	160	235
,, two ,, compound,	$15 \times \sqrt{p}$	$13 \times \sqrt{p}$	$18 \times \sqrt{p}$
Two-crank ,, ,, expansive,	14 ,,	12·2 ,,	17 ,,
,, ,, ,, compound,	10 ,,	8·5 ,,	12 ,,
Three-crank three ,, ,,	8 ,,	7·0 ,,	9·8 ,,
,, triple expansion, - -	6·2 ,,	5·2 ,,	7·3 ,,
Four-crank triple or quadruple, - -	6·6 ,,	5·8 ,,	8·0 ,,
Naval engines, three-crank, - -	72	63	100
,, four- ,, - -	75	61	75

The crank- and screw-shafts of ships which are run at full speed only occasionally, and for short periods, may be calculated by taking F at half the above values.

Example.—To find the diameter of the crank-shaft for a three-crank triple-compound engine having cylinders 30 inches, 45 inches, and 75 inches diameter, and a stroke of 50 inches. The load on safety valve is 165 lbs., the revolutions 90, and the I.H.P. 3300.

(1) By above Rough Rule.

$$\text{Diameter of crank-shaft} = \sqrt[3]{\frac{3300}{90} \times 6{\cdot}2\sqrt{165}} = 14{\cdot}3 \text{ ins.}$$

(2) By Board of Trade Rule.

$$\text{Diameter of crank-shaft} = \sqrt[3]{\frac{25 \times 180 \times 75^2}{1110\left(2 + \dfrac{75^2}{30^2}\right)}} = 14{\cdot}0 \text{ ins.}$$

(3) By Lloyd's Rules.

Diameter of crank-shaft

$$= \frac{21}{20}(\cdot038 \times 30 + \cdot009 \times 45 + \cdot002 \times 75 + \cdot0165 \times 50)\sqrt[3]{165} = 14{\cdot}5 \text{ ins.}$$

(4) By British Corporation Rule.

$$\text{Diameter of crank-shaft} = \sqrt[3]{\frac{180 \times 75^2 \times 50}{17{,}300}} = 14{\cdot}3 \text{ ins.}$$

(5) By the Rules of the Bureau Veritas.

$$\text{Diameter of shaft} = \sqrt[3]{\frac{165 \times 50\,(30^2 + 0{\cdot}1 \times 75^2)}{3900}} = 14{\cdot}56 \text{ ins. (iron),}$$

TABLE XVIIIa.—Shafts for Paddle Engines.

Description of Engine.	Value of F Intermediate Shaft Journal.	Value of F Paddle-shaft Inner Journal.	Value of F Paddle-shaft Outer Journal.
Single-crank single-cylinder,	...	80	100
Two-crank two-cylinder; cranks virtually at right angles, and connected by link,	...	58	65
Two-crank two-cylinder, with intermediate shaft; cranks at right angles,	58	50	65
Two-crank two-cylinder; solid crank-shaft; cranks at right angles,	...	55	65

For paddle steamers working only in smooth water the above values of F may be reduced 20 per cent.

The above values of F will suffice for iron shafts, but it is better not to reduce them when employing mild steel, unless the forgings are from ingots of the highest quality.

Details of Crank-shafts.

Crank-arms if forged solid with the shaft—

*Breadth - - - - = 1·1 × diameter of shaft.
Thickness - - - = 0·75 × „
Diameter of coupling - = 2·0 × „
Thickness „ - - = 0·3 × „
Number of coupling bolts = 2 + diameter of shaft in inches ÷ 2.
Diameter „ „ = diameter of shaft ÷ $\frac{4n + 19}{10}$.
Diameter of crank-pins - = 1 to 1·1 × diameter of shaft.
Length „ - = 1 to 1½ the diameter of shaft.
Length of journals - - = 1 to 1½ „

The following rules give sizes closely approximating to those found in practice, and may be used to obtain the diameter of the shafts, preliminary to making a more elaborate calculation. They are, therefore, very useful in the initial stages of a marine engine design to enable the designer to get on with the work; but being purely empirical they should be used with some caution:—

d is the diameter of the H.P. cylinder in inches.
d_m „ „ M.P. „
D „ „ L.P. „
S the stroke also in inches.
F a factor, which for the crank-shaft of ordinary compound engines with cranks at 90° is 12; and for the tunnel-shafts is 13; for three-crank triple-compound engines F is 14 for the crank-shaft and 15·2 for the tunnel-shafts.

(1) Ordinary compound engines

$$\text{Diameter of shaft} = \frac{d + D + S}{F}.$$

---

* $\text{Breadth}^2 \times \text{thickness} = 0{\cdot}9 \times d^3$.

(2) Triple-compound three-crank engines

$$\text{Diameter of shaft} = \frac{d + d_m + \text{D} + \text{S}}{\text{F}}.$$

Since it is usual for $d + d_m$ to equal or nearly equal D; and d_m is usually equal to $1{\cdot}5\,d$; then $d + d_m + \text{D} = 2\,\text{D}$ or $5\,d$. Then diameter of crank-shaft of a three-crank triple-compound engine $= \frac{5d + \text{S}}{14}$ or $= \frac{2\text{D} + \text{S}}{14}$.

Hollow Shafts.—The steel shafts in naval ships are now almost invariably made hollow in order to have the maximum strength with the minimum weight. It is seen that the strength of shafts varies as the cubes of their diameter, but the weight varies as the square; hence, a shaft of 10 inches diameter is only 23½ per cent. heavier than one of 9 inches diameter, but by the above rule it is 37 per cent. stronger. Now, if the weight of the 10-inch shaft be reduced to that of the 9-inch shaft by boring a hole 4·36 inches through it longitudinally, its strength to that of the latter will then be roughly $10^3 - 4{\cdot}36^3$ to 9^3; or 917 to 729, or nearly 26 per cent. stronger.

This hollow shaft will also be much stiffer against bending by its own weight, and therefore very suitable where the supports are necessarily far apart; it will, however, be much more costly of manufacture than a solid shaft.

The exact calculations for hollow shafts differ somewhat from that given before on the basis of the cubes of diameters; the late Professor Rankine showed (*Applied Mechanics*, p. 355) that if d is the diameter of a solid shaft, d_1 that of a hollow one whose internal diameter is d_0, then

$$\frac{f \times d^3}{5{\cdot}1} = \frac{f(d_1^4 - d_0^4)}{5{\cdot}1 \times d_1}; \text{ or } d^3 = \frac{d_1^4 - d_0^4}{d_1}.$$

Now, in general practice, $d_0 = \frac{d_1}{2}$.

Substituting this value in the above,

$$d^3 = d_1^3 \times \frac{15}{16}; \text{ or } d_1 = d\sqrt[3]{\frac{16}{15}} = d \times 1{\cdot}022.$$

That is, the diameter of a hollow shaft to equal a solid one in strength should be only 2 per cent. larger when the hole is as much as half the diameter.

CHAPTER X.

FOUNDATIONS, BED-PLATES, COLUMNS, GUIDES, AND FRAMING.

For the good working of an engine it is essential that the fixed parts, such as bed-plate, framing, &c., shall not only be strong enough to resist the strains to which they are subject, but rigid and stiff enough to prevent any tendency to change of form which would throw abnormal stresses on the working parts. With this object in view it is usual to construct such parts of cast iron or cast steel; and from their form and general construction this enables the engineer to manufacture a cheap structure having the necessary qualities. But since cast-iron has not a high tensile strength, and is comparatively unsuited to withstand sudden shocks, structures made of it cannot be so light as if made of wrought-iron or steel; so that when extreme lightness of machinery is aimed at, the framing is usually made of forged steel or wrought iron, and rigidity given to it by cross bracing, &c. This latter system is, of course, an expensive one in most engines, and only adopted when economy of weight is of more importance than economy of manufacture. Although cast-iron framing and bed-plates are undoubtedly cheaper and better for engines generally, and continue to be used for all sizes in the mercantile marine, a system of construction with wrought-iron or steel for very large engines may be adopted with advantage, and taking into account cost of patterns and risk in casting may be more economical.

Steel manufacturers can now, however, produce large and moderately complicated castings in steel, and the foundations and frames of naval engines of large sizes are being made wholly of that material. Increased experience in the making of steel castings will no doubt give greater faith in their soundness as well as permit of a lower price, when no doubt they will be more freely used for columns and foundations (fig. 12). At present they can only be used to advantage in large engines owing to the great thickness demanded by the steel moulders.

Bed-plates and Foundations.—Vertical engines are usually built on a structure called by these names, and sometimes known as the *sole-plate*. It contains the main bearings for the crank-shaft, and on it are the facings for the columns, condenser, &c., and it sometimes contains the waterways leading from the condenser to the pumps. This form of sole-plate underlies the *whole* of the engine, and is most suitable when the engines are to be fitted into a ship of light construction; it is, however, somewhat heavier and more expensive than the one generally adopted (fig. 21), when the condenser is fitted with horizontal tubes in a fore and aft direction. The foundation then contains only the main bearings, and has facings for the front columns only; it is bolted to feet on the front of the condenser, so that with the latter it forms the base or engine superstructure. The condenser is in this case lower down, and the weight and cost of half the sole-plate is saved. The part of the foundation fitting to the feet on the condenser front should be of good depth, the flanges strong, notched one into the other, and strongly bolted at top and bottom. The transverse parts of the bed-plate, into which the main bearing brasses are fitted, are sometimes formed like inverted bowstring girders, and are unsupported by the bed built in the ship, but span the space between the fore and aft bearers. This is convenient sometimes, especially when the shaft must be low down in ships with a good rise of floor, and also for very small engines; but it must be of strong form, and as it depends for strength only on the connection to the longitudinal parts of the foundation, it is a most convenient form when steel castings are used, as also for very light engines; additional support may be, and often is, given in the case of large engines by bolting to the ship at the middle (fig. 66). When this particular style is adopted great care should be exercised in designing the foundation, so that the transverse portions have a good extended connection to the longitudinal ones, especially in the direction of the column bases. If the longitudinal parts are flat and straight on the bottom, so as to be in the same plane as the rest of the foundation, they may be formed with flanges and bolted to the steel seating in the ship, and from it receive support and strength.

Main Bearings.—The bearings in which the shaft journals run should approximate, as far as possible, to a hole through a solid support. If it were practicable a hole with a bush of suitable metal in it would form the best possible bearing for a shaft; but since the bearing, however well designed and made, is liable, in course of time, to wear somewhat, it becomes a necessity that there shall be some means of adjusting the brasses, so as to prevent the shaft from having side movement when they are worn. In the case of the vertical engine, the weight of the shaft, and the pressure from the piston, act very nearly in the same direction, so that the wear is only vertically above and below the shaft; consequently the adjustment is necessary only in a vertical direction. The greatest strains on the bearings, however, are during the first

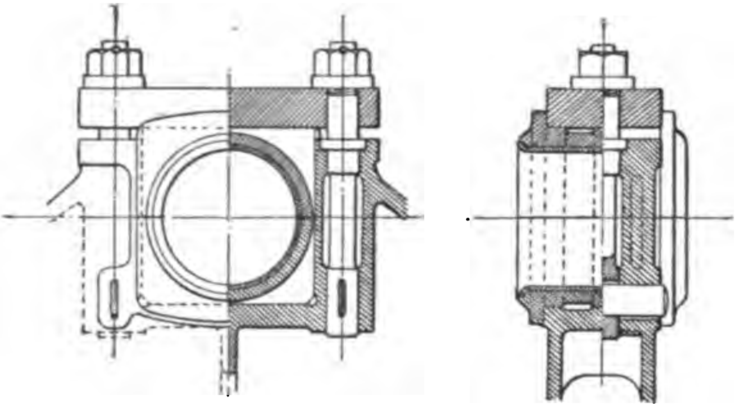


Fig. 63.—Crank-shaft Bearing.

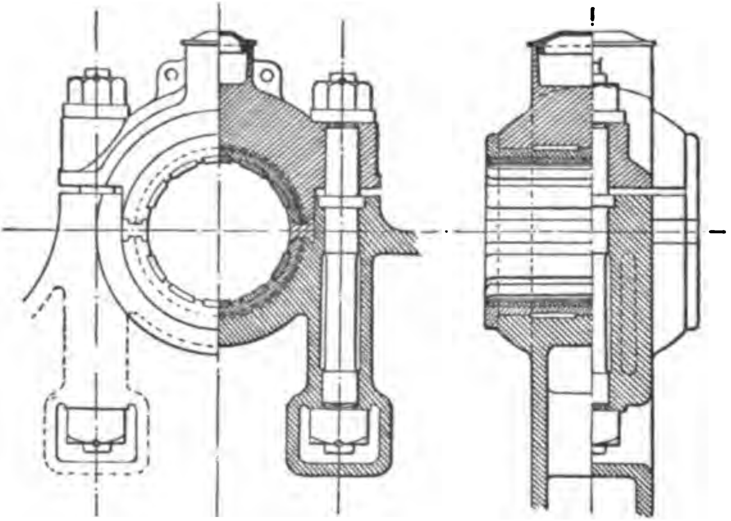


Fig. 64.—Improved Form of Crank-shaft Bearing.

half of the stroke, and consequently the position of mean pressure on the journals is not exactly vertical; this is also somewhat modified on the upstroke by the tendency of the shaft to roll on the surface of the brasses, and on the downstroke it is aggravated from the same cause. In fitting the brasses for a vertical engine, this should be borne in mind, and every allowance made for taking the wear due to these causes. It is of the utmost importance for the good working and endurance of a crank-shaft, that the bearings be rigid in themselves, and that the framework containing them shall be rigid enough to sustain them perfectly in line one with another. Crank-shafts are more severely tried by the giving or springing of the bearings than any other cause, and they were oftener broken from want of rigidity in the bed-plate and seatings, than from the normal strains from the pistons, so that a shaft may be of ample size to bear the twisting and bending strains if properly supported in its bearings, and yet give way after a few weeks' work in a light foundation in a weak ship.

The brasses are usually formed as shown in fig. 63, and carefully bedded into the recesses provided for them in the foundation. At one time it was usual to design them with projecting facings, called chipping strips, to avoid the labour of chipping and filing the whole of the surface; this was, however, found to be highly objectionable as engines increased in size, and with the increase of boiler pressure, and consequent increased percussive action due to the high initial loads, such an effect was produced on these strips and the cast-iron surface on which they were borne, that engineers have gradually abandoned the practice; the planing machines also have rendered such a device unnecessary, as it is nearly as cheap to fit brasses so as to bear over the whole surface as to do so only on strips. The square bottom brass is objectionable on two grounds; one being that it is impossible to remove it in most engines without lifting the shaft, and the other that when it becomes hot it is invariably distorted, from its variation in thickness of metal, with the ultimate result that it is broken through the middle longitudinally.

The first of these evils is avoided by making the bottom brass round and of even thickness, so that it can be got out when relieved of the weight of the shaft, by being moved around until it is on the top of the journal. The second evil is also partly avoided by making it of an even thickness; but this form of brass is often found cracked, and is liable to heat from its want of stiffness. Both these brasses, when first heated by abnormal friction, tend to expand along the surface in contact with the shaft; this would open the brass, and make the bore of larger diameter, if not prevented by the cooler part near the cast iron, and by the bed-plate itself. If the brass has become hot quickly and excessively, the resistance to expansion produces permanent set on the layers of metal near the journal, so that on cooling this contracts, the brass closes and tends to grip the shaft; it will then set up sufficient friction to heat again, and expand sufficiently to ease itself from the shaft, and so long as that temperature is maintained the shaft runs easily in the bearing. This is why some bearings always are a trifle warm, and will not work cool. A continuance of heating and cooling will set up a mechanical action at the middle of the brass, which must end in rupturing it, just as a piece of sheet metal is broken by continually bending backwards and forwards about a certain line.

This action of the brass can be prevented by securing it to the bed-plate, along its two longitudinal edges, as shown in fig. 64, by an H shaped strip, which holds both top and bottom brasses, so that they cannot move in their beds. This method is a very simple one, and has been most successful in engines of all sizes.

It is also essential that the bearing to be efficient should be rigid throughout its whole length; this is not the case when the brasses have long overhanging ends, which afford little or no support to the shaft. To this end it is better, when possible, to extend the bed for the brasses, so as to support them over the whole of their length, as shown in fig. 64.

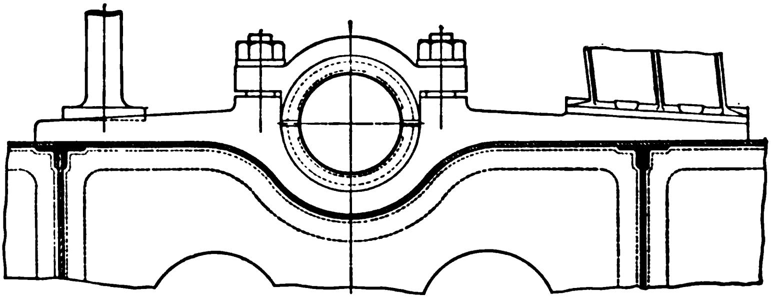


Fig. 65.—Solid Steel Main Bearing Frame.

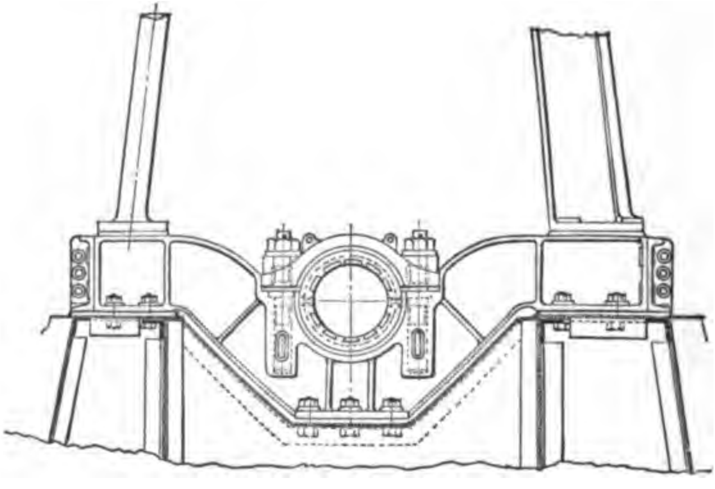


Fig. 66.—Cast-steel Main Bearing Frame for Naval Engines.

Caps or Keeps for Main Bearings are very generally made of wrought iron, but as stiffness is as necessary as strength, cast steel, as in fig. 65, or even cast iron, as in fig. 64, may be used with advantage in their construction. A wrought-iron cap, which may be amply strong, is often far from stiff enough, while a cast-iron cap, which is stiff enough for good working, is generally amply strong.

Let *d* be the diameter of the main-bearing bolts (when there are only two to each cap), *t* the thickness of the cap, and *b* its breadth, *l* is the pitch of the bolts, all in inches; f a factor, which for wrought iron and forged steel is 1, for cast steel 1·2, and for cast iron 2.

$$\text{Thickness of cap} \quad = d\sqrt{\frac{l}{b} \times f}.$$

$$\text{Thickness of brass at middle} = \frac{d}{3}\sqrt{\frac{l}{b}}.$$

Main-bearing Bolts.—Each cap is usually held by two bolts, but very large bearings have four bolts, two on each side, so as to avoid large bolts and heavy nuts, and to distribute the strain over the cap. When everything is in good order and properly adjusted, the load from the piston should be equally divided between the bolts; but since, from a very slight difference in setting of the nuts, the load may come on three, and sometimes even on two bolts only, due allowance must be made for this. To meet this it should be assumed that each bolt is capable of sustaining one-third the load on the piston. If P is the maximum load on the piston in lbs.,

$$\text{Area of each bolt at bottom of thread} = \frac{P}{3f}.$$

For mild steel f = 5000 lbs. for small and 7000 lbs. for large bolts.
For good iron f = 4500 „ „ 6000 „ „ (*v.* Table xlvi *c*).

Also,

$$\text{Diameter of main-bearing bolt} = \text{diameter of cylinder} \times \sqrt{\frac{p}{3f}}.$$

p is the maximum pressure per square inch, and is, as stated on p. 164, and may be taken generally at 0·75 × absolute boiler pressure for the high-pressure cylinder of a triple engine, to equal the load on the low-pressure piston.

Brasses, so called because generally made of brass.—They should be made of a metal which will withstand wear without wearing the shaft journals, and whose surface is such that the shaft runs on it with a minimum amount of friction. The metal must also possess sufficient strength not to fracture under the percussive strains of the piston, and be free from brittleness, so as not to crack when quickly cooled. Good gun-metal or bronze possesses *all* the qualities essential for brasses when the journal is of wrought iron or fairly hard steel; but the soft steel as used for shafts will not run satisfactorily on bronze bearings. There are, however, other metals which have certain of these qualities in a higher degree without having them all. Cast iron is harder than ordinary bearing bronze, and when once worn to a smooth surface gives equally good results; but it is liable to fracture from continued shocks and when cooled suddenly. White metals offer least resistance, or produce least friction, but most of them are too soft to be used alone. Of the patent bronzes there are few which are suitable for heavy bearings, and none of them have so far been shown to be much superior to good gun-metal.

White metal carried in a bronze or cast-iron shell is beyond doubt the best bearing for both wrought-iron and soft steel journals, and if kept well lubricated there is practically no wear.

When a bearing is of ample size, properly designed and constructed, well lubricated and looked after, it may be of almost any kind of white metal. If the bearing surface is limited there is a great difference in the behaviour of different metals; and if badly designed and constructed even the best metal will give trouble; but if not properly looked after by the engineer, the best metal and the most careful design are of no avail.

Certain of the white metals have so far given the best results as a bearing surface, and there is every reason for this, inasmuch as they do not cause abrasion of the shaft, and if their own surface is injured it will not, as a rule, form into fine sand, and grind both the surfaces, as all the bronzes do more or less. When white metal is used it is highly important that the shaft shall bear wholly on it, and not partly on it and partly on the metal containing it, and also that efficient courses for the distribution of the lubricant are provided.

There are three common methods of fitting the white metal into a setting of other metal—(1) By casting it into oblong recesses; (2) by casting it into a large number of small circular recesses; and (3) by driving in strips into longitudinal grooves, in the same way as the lignum vitæ in a stern bush. The last plan is, on the whole, the most satisfactory, for the strips are well secured, and extend over the whole length of the bearing, leaving several oil courses longitudinally, and the shaft bears on the white metal only; this also possesses the advantage that a strip may be taken out, and a new one refitted with ease, and without heating the brass and running the risk of distorting it. Cast iron is now very often used as a setting for the white metal, and answers the purpose very well indeed, being much harder than brass, and thereby better supporting the softer metal. When cast iron is used it is generally made thicker than when of brass; and sometimes advantage is taken of this to cast the shell hollow, so as to admit of its being filled with water.

The thickness of "brasses" in the crown depends principally on the diameter of journal.

When of bronze - - - - - = 0·10 × diam. of journal + 0·10 in.
„ cast iron - - - - = 0·15 × „ + 0·15 in.
When fitted with white metal thickness = 0·20 × diameter + 0·15 in.

When fitted with white metal strips as follows:—

Thickness of strips - - - = 0·04 × diam. of journal + $\frac{1}{8}$ in.
Breadth „ - - - = 0·16 × „ + $\frac{1}{2}$ „
Space between strips - - - = thickness of strip.
Thickness of metal beyond - - = 0·065 × diam. of journal when brass.
„ „ „ - - = 0·12 × „ „ iron.

Columns.—The columns which support the cylinders of a vertical engine are subject to alternate tensile and compressive stresses from the steam pressure in the cylinder; to a steady compressive stress from the weight of the cylinders, &c.; and to cross-breaking stresses when the ship is rolling and pitching. As a rule, columns, if designed from considerations of strength only, would not be stiff enough for good working. The same reasons, therefore, which decide the using of cast iron for foundations strongly influence most engineers to choose this metal for columns. The front columns are often made, however, of wrought iron or steel, turned smooth (*vide* fig. 66); and all the columns for exceedingly light engines are made of steel or wrought iron, well braced together to prevent vibration (*vide* fig. 13). Since cast iron is so superior to wrought iron or steel for resisting compression, and so inferior to either for resisting tension, a good composite column is formed by fitting a wrought-iron or steel tie-bar through a hollow cast-iron column, the latter supporting the cylinder while the former holds it down. The chief objection, however, to both this composite column and those of wrought iron and steel for large engines is the difficulty of getting good attachment to the cylinder; and since it must be outside the cylinder the columns are necessarily far apart, and away from the centre line of compound engines. To avoid this difficulty wrought-iron and steel columns are formed with a flange at each end like a shaft-coupling; but even then the strain on the cylinder is very much concentrated. Columns when of cast iron or of cast steel (*v.* fig. 12) are made of various shapes, and no rule can be laid down in favour of any particular form.

The columns should be so arranged as to support the cylinders and resist the reaction on the foundation. Some engineers, in thoroughly effecting the latter, completely hide from view the working parts, and make all the bearings, &c., very inaccessible (*v.* fig. 11). They should be so placed at the cylinder bottoms that the piston-rod centre is within the lines drawn through the extreme points of back and front columns; and when there are only two columns to each cylinder, the front ones are better to be spread out somewhat, so as to act as struts to resist any tendency to motion of the cylinders when the ship is rolling and pitching, and so as to leave the working parts more open to view. When there are guides on both back and front columns, or when the front columns only have the guides, then this is not possible.

Back columns are generally of different form from the front ones, to suit the guides and bearings for the pump weigh-shaft or levers when so fitted.

Some engineers utilise the back columns as exhaust pipes to the condenser, but this is not good practice, inasmuch as the heat of the steam causes them to expand, and when the guides for the piston-rods are on them the heat is conducted to them with prejudicial results; this latter difficulty is sometimes overcome by placing the guides on the front columns. It is also bad practice to expose any important part of the engine structure subject to heavy strain to unnecessary wear, such as wasting of the inner surfaces of the casting.

Guide-plates.—In order to have a sound and hard surface for the piston-rod slides or shoes to work on, the guide-plates should be separate from and secured to the columns; when so fitted they also admit of adjustment, and may be cast hollow, so as to permit of a flow of cooling water through them. This is especially needful for large quick-running engines, where the speed of piston is very high, and any casual want of lubrication would soon cause most serious damage. Cast iron when once worn smooth gives a splendid surface for a slide, but if by any mischance this surface suffers a little abrasion, it is most difficult to get right again, and will seldom work well again until it is replaned, &c.

The face of the guide plates should have good oil courses cut on it, so that the lubricant is well distributed, and they should be cut deep enough to prevent them being choked with the gluey deposit from the oil. The piston-rod slide should always be provided with a comb, which will carry the lubricant from the drip-boxes, and spread it over the face of the guide, which may with advantage have recesses to catch the oil so spread, where it is retained and stored to trickle down at leisure.

Framing.—Horizontal engines require a different arrangement of bed-plate and framing from that of the vertical type. Trunk and return connecting-rod engines have no sole-plate proper, as the cylinders are connected to the condenser casting (fig. 14) by **A** frames, which contain the main bearings; the trunk engine required no guides, and those for the crossheads of the return connecting-rod engine are on each side of the condenser. These frames must be sufficiently strong to take the whole strain from the pistons, and stiff enough to remain rigid under those strains, or the crankshaft will be liable to distortion. The usual form approximates to the letter **A**, the two feet being connected to the cylinder front, one at top and one at bottom, in line with the brackets on which the cylinder sits, and by which it is secured to the seatings in the ship. Projecting feet should be cast on the cylinder front to meet those of the frames, so that the connection may be made with driven bolts.

Side stability is obtained for the frames by splaying out their feet sideways, and by making them with a broad base, well stiffened by webs and fillets.

Usually there were only three frames to a two-cylinder engine, and four frames to a three-cylinder engine, the middle ones being very much stronger than the other two, as they were required to take part of the strain of both engines.

The brasses should be fitted so that the centre line through them, instead of being horizontal in the transverse plane, should be at an angle with the horizontal line, whose tangent is equal to the weight of half the shaft, divided by the mean pressure on the crank-pin; and for convenience of fitting in and removal of the shaft, some engineers incline them more than is given by this rule. There was of necessity a somewhat weak connection between the frames and the condenser casting, and although this was not of serious consequence in the trunk engine, it was often a cause of trouble in the return connecting-rod engine, and every care should be taken to make it as secure and rigid as possible, and the utmost pains taken to have a strong and rigid seating under the frames, which shall so stiffen the ship and engine as to prevent the crosshead guides and piston-rods from getting out of line with one another.

Horizontal direct-acting engines have a sole-plate, which connects the cylinders to the condenser casting, and contains the guides for the piston-rods, and the brackets for the main-bearings; these latter are usually stayed to the cylinder tops by tie-bars through cast-iron struts or by steel tie-rods only.

The framing for diagonal paddle-wheel engines is made somewhat on the same principle as that for the horizontal screw-engine, with modifications (fig. 7) to suit the altered conditions. The main part of these frames must extend from the cylinder to shaft and down again, so as to form a support for the latter, and having guides for the piston-rod crossheads. Intermediate supports or columns connect this main frame to the foundation. It is now usual to make these frames of wrought-iron or steel bars (*vide* fig. 9), and for some very light draught steamers of large power frames made of steel angles and plates have been found considerably lighter than the ordinary cast-iron frames, and can be designed to add materially to the stiffness and strength of the ship in the neighbourhood of the machinery. A modified form of this kind of frame is imperative, when exceedingly light draught is a necessity, as the hull is so light, to comply with the requirements, as to be unable by itself to stand the racking strains from the engines.

Even when weight is of secondary consideration, the wrought-iron or steel frame is preferable to the cast iron, and when the cost of patterns is taken into account, it is no more expensive. The bearings for the shaft, and the guide-plates for crossheads, are, of course, of cast iron or cast steel fitted to the wrought-iron work direct.

Side stiffness and stability are obtained by connecting the four frames by wrought-iron tie-bars through the top (fig. 8), and by the main beam before the shaft. The main-bearings for the shaft of a diagonal engine are sometimes so set that the shaft can be lifted vertically; but a better plan is to set them at a slight angle, so that their centre line is in the direction of the *resultant* of the *weight* of shaft, &c., and *mean pressure on the journals* due to the thrust on the connecting-rod.

Entablature of Oscillating and Steeple Engines.—This is usually of cast iron, but may with advantage be made of wrought iron, or steel plates and angles, or, better still, of cast steel, as the strains on it from the overhung cranks are severe and concentrated owing to its being supported at so few points. It was no uncommon thing to find it broken and patched after a few weeks' work, and very few of them work without a certain amount of vibration, which must tend to produce rupture in course of time.

It is usually supported (fig. 6) by four columns to each crank, and additional stiffness and stability are imparted by diagonal cross braces to the foundation at each end. It is seldom possible to place the supporting columns in line with the main girders of the entablature, but when possible this should always be done, so as to avoid the canting action which is caused by the centre of support not being in the same plane with the centre of force on the journals. When this is not possible, the sides of the entablature connecting the main girders or rockers should be of extra stiffness, and well connected to them by spreading out webs or fillets. Special advantage should also be taken of the main beams of the ship, to form a powerful tie to the entablature girders, and to prevent their tendency to canting. This can be done generally by multiplying the number of the bolts, and fitting cast-iron filling pieces in lieu of hardwood ones only. The bottoms of the cross-bearers should, when possible, be tied together by bars parallel with the crank-shaft to keep them from "giving" or twisting. In the case of steeple engines the thrust of the connecting-rod on the guides caused great stress on the entablature and framing, and necessitated diagonal ties between it and the foundation. The horizontal force in an oscillating engine also puts cross strains on the entablature which the ordinary columns are not fitted to resist, hence the diagonal cross frame which Penn generally fitted.

To resist, as far as possible, the tendency to spring, the supporting columns should be of extra size, with strong and broad flanges.

When there are four supporting columns of wrought iron or steel, their diameter in the body should be 0·7 the diameter of the piston-rod, and at each end 0·55.

The collars or flanges on which the entablature is carried, should be equal in diameter to that of the piston-rod, and 0·2 the diameter of the piston-rod in thickness.

If the columns happen to come in line, or nearly so, with the centre of shaft journal, they may be 10 per cent. less in diameter than given above. The breadth of the rockers should be not less than the diameter of the shaft journals, and the depth at the centre should be calculated as for a box-girder, subject to sudden loads applied at the middle of its length, which is measured from column to column.

Roughly speaking, the depth of the rocker under the bearing brass should not be less than four times the diameter of the piston-rod for engines of ordinary dimensions.

The thickness of metal of sides of rockers

$$= 0{\cdot}4 \sqrt{\text{diameter of piston-rod.}}$$

Thickness of top and bottom $= 0{\cdot}6 \sqrt{\text{diameter of piston-rod.}}$

The bottom brasses of the main bearings should be round, so that the recesses for them may tend to strengthen the rockers rather than weaken them, as would be the case if square-bottomed.

CHAPTER XI.

THE CONDENSER.

THE function of the condenser is so to cool down the exhaust steam as to reduce its pressure to a minimum, and in doing this the steam is converted into water. The very early engines could only work by the aid of condensation, as the steam with which they were supplied was generally of a lower pressure than the atmosphere; it is, in fact, owing to this that the steam-engine owes its birth, for steam was preferred by the early mechanicians because it was so readily changed from a gas to a liquid, and so produced that vacuum which Nature was supposed to abhor, and to fill which she would do the work of horses. The exact relation of the condenser is better understood by following the early history of the steam-engine from the day when the cooling water was admitted to the cylinder after the steam, and then allowed to run freely away from the bottom on the descent of the piston, to the time when Watt, having perceived the waste of energy in always forcing the piston up against the atmospheric pressure, and in admitting the hot steam into the cold cylinder, made the engine double acting, and effected the condensation in a separate chamber. The jet of water continued long after Watt's time as the means of cooling the steam, and gave in later days the distinguishing name to the condenser, which is now nearly entirely superseded by a more perfect apparatus.

The Common or Jet Condenser, now really *uncommon*, consists essentially of an air-tight chamber, into which the steam flows from the cylinder after having been exhausted of its available energy; the passage of the steam is intercepted by a spray of water, caused by the inrush of water through small holes or narrow slits in a pipe placed across the steamway. If the spray is fine, like a shower of rain, it mixes mechanically with the steam, as well as cools it by surface contact; should the pipe have slits, so as to cause the water to flow in thin broad streams like ribbons, the cooling is principally effected by surface contact. The result in either case is the turning of the steam into water, which falls to the bottom, and is pumped away by the *air-pump*. It might be supposed that the mere turning of the steam into water, thereby causing it to occupy far less space, will cause the vacuum in the condenser; it does, but to so slight an extent and of such an evanescent nature, that unless some other means were at hand, the condenser would be useless, Water readily absorbs air when freely exposed to the atmosphere, and gives it up again on being heated. The feed-water contains air, which becomes mechanically mixed with the steam in the boiler, and passes with it through its various passages, until it enters the condenser, when it parts company with it, and remains as cooled air after the steam is converted to water. The cooling water also contains air, and readily gives some of it up on becoming heated by the exhaust steam, especially under the diminished pressure of the condenser. After a few strokes of the piston a sufficient amount of air will be accumulated to raise the pressure to that of the atmosphere, and although the condenser may be kept quite cool there will be no vacuum, but the reverse. The pump, therefore, which exhausts the condenser draws away the air as well as the water, but since the latter could run away by gravity, it is only the former which is of necessity pumped; hence this pump is rightly named the *air-pump*.

The *shape* of a jet condenser is immaterial so long as the inlet for the steam is high enough to prevent the water running back into the cylinder, and the bottom so formed that the water will all drain into the air-pump bottom. It is generally formed to suit the ship and the working parts of the engine, and was often a part of the engine framing; the back columns of vertical engines were often utilised for the purpose, and did extremely well, except that occasionally rapid corrosion so weakened them as to become dangerous. The frames of the horizontal engines were arranged by some engineers to do duty for condenser, until the Admiralty finally forbade the practice.

The *capacity* of the jet condenser should not be less than one-fourth that of the cylinder or cylinders exhausting into it, and need not be more than one-half of it, unless the engine is a very quick running one; one-third the capacity is generally, however, sufficient. The objection to a large condenser, beyond its cost and weight, is that a longer time is necessary to form a good vacuum in it; and the objection to a small condenser is its liability to flooding and overflowing to the cylinders, unless the engineer is most attentive.

The amount of injection water depends on the *weight* of steam to be condensed *and its temperature;* the *exact* quantity of water per pound of steam depends on the temperatures of the steam, of the cooling water, and of the "hot-well," or receptacle into which the air-pump delivers the products of the condenser. As the supply of water to the boilers (called the *feed-water*) is taken from the hot-well, and it is an obvious advantage for it to be as warm as possible, the cooling water used is only such as sufficient to produce a good vacuum. With the jet condenser a vacuum of 24 inches was considered fairly good, and 25 inches as much as was possible with most condensers; the temperature corresponding to 24 inches vacuum, or 3 lbs. pressure absolute, is 140°. In actual practice the temperature in the hot-well varies from 110° to 120°, and occasionally as much as 130° is maintained by a careful engineer. To find the quantity of injection water per lb. of steam to be condensed:

Let T_1 be the temperature of the steam whose latent heat is L; T_0 the temperature of the cooling water, whose quantity in lbs. is Q; T_2 the temperature after condensation, or that of the hot-well.

The *total heat* of the steam $= T_1 + L$.

The heat absorbed by the cooling water will be $(T_1 + L) - T_2$.

But the heat absorbed by the cooling water is also represented by $Q\,(T_2 - T_0)$. Hence

$$(T_1 + L) - T_2 = Q (T_2 - T_0).$$

Or

$$Q = (T_1 + L) - T_2 \div (T_2 - T_0).$$

Now $(T_1 + L) - T_2$ is equivalent to the total heat of evaporation from T_2 and at T_1, and is therefore equal to $966^\circ + 0{\cdot}7 \times 212^\circ + 0{\cdot}3 \times T_1 - T_2$.
Or

$$Q (T_2 - T_0) = 1114^\circ + 0{\cdot}3 \times T_1 - T_2.$$

Therefore

$$Q = \frac{1114^\circ + 0{\cdot}3 \times T_1 - T_2}{T_2 - T_0}.$$

Example.—To find the amount of injection water required for an engine, the steam at exhaust being at a pressure of 10 lbs. absolute; the temperature of the sea is 60°, and it is required to keep the hot-well at 120°.

The temperature corresponding to 10 lbs. is 193°.

$$Q = \frac{1114^\circ + 0{\cdot}3 \times 193^\circ - 120^\circ}{120^\circ - 60^\circ} = 17{\cdot}53 \text{ lbs.}$$

That is, the amount of injection water is 17·53 times the weight of steam for this particular case.

The allowance made for the injection water of engines working in the temperate zone is usually 27 to 30 times the weight of steam, and for the tropics 30 to 35 times; 30 times is sufficient for ships which are occasionally in the tropics, and this is what was usual to allow for general traders.

The Area of injection orifice and size of pipes is governed by the head of water, vacuum, and length of piping, or, in other words, by the *equivalent head* at the condenser.

Neglecting the resistance to flow at the orifice, and in the pipes and passages, the velocity at the condenser may be found as follows:—

Let h be the head of water above the valve on the condenser in feet, p the pressure in the condenser, and h_1 the equivalent head g for gravity,

$$h_1 = h + (15 - p)\, 2{\cdot}3,$$

and velocity in feet per second $= \sqrt{2 g h_1} = 8{\cdot}025 \sqrt{h_1}$.

Example.—To find the theoretical velocity of flow into a condenser in which the vacuum is 26 inches, and the orifice 12 feet below the water-line of the ship. Here

$$h_1 = 12 + (15 - 2)\, 2{\cdot}3 = 42 \text{ feet.}$$

Then

$$\text{velocity} = 8{\cdot}025 \sqrt{42} = 52 \text{ feet per second.}$$

In practice owing to loss of head due to resistance at valves in the pipes, &c., the actual velocity is only about half that given by the above rule. Hence, in designing it is usual to calculate on a velocity of only 25 feet per second for shallow draught steamers, and 30 for deeper ones.

From these rules, and with such allowances, the following holds good:—

Area of injection orifice in square inches = number of cubic feet of injection water per minute ÷ 10·4 to 12·5 according to circumstances, or = weight of injection water in pounds per minute ÷ 650 to 780.

A rough rule sometimes used is—

Allow one-fifteenth of a square inch for every cubic foot of water condensed per hour. And another

Area of injection orifice = area of piston ÷ 250.

The injection valve is usually a simple slide or sluice valve, which is readily opened or shut, and regulates the amount of water; for large engines the valve should be of the gridiron type, to avoid large travel. The handle or lever for working the injection valve must be very near the starting gear, so that the water may be shut off as soon as the engine stops.

Snifting Valve.—It is usual to fit, near the bottom of the condenser, a non-return valve, through which the water, &c., may run or be blown out by steam; it shuts by its own weight, and is pressed on its seat by the pressure of the atmosphere. This is called the *snifting* valve, and it allows of the condenser being emptied of water and air by steam before starting the engine, and likewise prevents the pressure in the condenser exceeding that of the atmosphere to such an extent as to be dangerous in case of mishap with the injection water. The valve through which the steam is admitted is called the *blow-through valve*, and was a simple mushroom valve, raised by means of a lever, and closed by the steam pressure on the lever being released.

The snifting valve was usually exposed without a casing, so as to be easily inspected or removed in case of being gagged with dirt, &c.; to prevent the water from being spurted about the engine-room, the valve was formed with a curved rim, which completely covered and overhung the seat like an inverted saucer.

The internal injection pipe or *rose* should be placed down below the flow of steam, so that the cooling water may pass twice through the steam.

Bilge Injection.—A second injection valve is fitted to the condenser, which admits water from the bilges, and is used in case of leakage of the hull. This was no uncommon occurrence in the old wooden paddle-boats and the full-powered screwships, so that this valve was often the means of saving the ship.

The area of this valve should be the same as that of the sea injection, but it was usually only about two-thirds of the latter; the reduction in area was partly due to the shorter pipes and straight leads, as compared with those of the sea injection.

The tail pipe of the bilge injection was either stopped at the end and had a number of small perforations, or was fitted with an efficient rose-box or basket.

Surface Condenser.—It has been seen that with jet condensation the contents of the hot-well consist of a mixture of sea water and condensed water in the proportion of about 30 to 1, so that the water available for feeding the boiler is very nearly as salt as sea water. If the cooling water is kept separate from the condensed steam, the latter, which is pure water, may be used as feed water. The idea is by no means a new one, since so far back as 1794 Cartwright took out a patent for an engine, in which the steam was condensed on the cold surfaces of two metal cylinders placed one within the other, and having cold water through the inner one and about the outer. An engine was made on this patent, and is said to have given great satisfaction. Brunel, in 1822, took out a patent for the same object; his invention was designed more especially for ships, and consisted of groups of small tubes. In 1833, L. Herbert and J. Don patented an arrangement whereby "the air and steam from the eduction passage is drawn by a fan through or among small tubes, so as to be condensed. The tubes may be below the water." In 1835, W. Symington patented a plan "for condensing the steam from the cylinder, and cooling the surplus water from the air-pump, by tubes laid along the keel exposed to water outside a steam vessel." In 1838, J. B. Humphreys took out a patent for "surface condensation by leading the steam through tubes in a vessel kept cold by a flow of water." In 1855, J. Biden claims as his invention a plan whereby "the steam is condensed by passing through tubes, which are led outside the vessel or through passages inside, kept cool by a flow of sea water, and the condensed fresh water is used for the boiler." It is even stated that James Watt, among many others, used a surface-condenser at one time; but the practical success of this mode of condensation is chiefly due to Mr. Samuel Hall, with whose name the surface-condenser is generally and properly associated. He took out a patent in 1831 for a system of surface-condensation, and in his specification claims, among other things, to condense the waste steam from the safety-valves, and to distil fresh water, to make up loss, by an apparatus, in principle similar to the distillers of to-day. One of the first ships fitted with Hall's condenser and appurtenances was the "Sirius," which made the first voyage under steam from England to America in 1838. In 1837 engines were made for the "Wilberforce," fitted with Hall's condenser, and it is interesting to note that these engines had piston valves, &c., pretty much as now used. It was not, however, till 1860 that the surface-condenser came into general use, and then only slowly; indeed, it was not until mineral oil was used exclusively for internal lubrication that troubles from surface-condensation ceased. In Hall's condensers the steam passed through the tubes and the circulating water outside them; had the cooling water gone through the tubes it is highly probable they would not have "choked with mud and sand" and been condemned, as they were in the case of the "Wilberforce" in 1841.

Condenser Tubes.—It is essential that the surface on which the steam is to be condensed should be metallic, because the material separating the steam from the cooling water must be thin and a good conductor of heat, strong enough to resist the pressure of the water on it, amounting to at least 15 lbs. per square inch, and capable of experiencing sudden changes of temperature without fracture.

The circular section being best suited to resist both internal and external pressures, tubes were naturally chosen as the means of separating the steam from the water, and these tubes admit of a very large amount of surface in a small space.

Copper, being one of the best conductors of heat, was at first chosen as the material from which to make the tubes, and being highly ductile, these could be drawn out very thin indeed. But it was soon found that the acids derived from the fatty matter from the cylinders dissolved some of the copper, and produced soluble salts of that metal, which were pumped into the boiler with the feed-water, and there caused great injury to the iron surfaces. This, for a time, gave the surface-condenser a bad repute, as it was found that the amount saved by them was exceeded by that representing wear and tear of the boilers. This was eventually obviated by having the copper tubes coated with tin, and by discontinuing the use of tallow in the cylinders. But, notwithstanding this, the boilers, especially those in H.M. Navy, showed signs of premature decay, such as was not customary with those receiving water from a jet-condenser. It was found to be due to the highly corrosive power of redistilled water on the bare surface of the iron, and to the impossibility of keeping a protective scale on the surfaces when such water was used; later research has shown also that some gases which entered the boiler with the original water *chemically combined* with bases, which kept them comparatively innocuous, were freed, and came back *mechanically mixed* with the feed-water from the hot-well, capable of highly destructive action on iron-surfaces.

Copper tubes were, of course, expensive, and the tinning added to their cost as much as 20 per cent., thereby rendering the first cost of a surface-condenser a considerable addition to that of the engine. In 1860 brass tubes had long been used for boilers, and as this material was very ductile, and its galvanic action on iron almost nothing, it was tried as a substitute for copper in the manufacture of condenser-tubes with at first mixed success, the want of complete success being partly due to want of care in manufacture, and partly to prejudice. The partial success has since become a complete success, and all condenser-tubes are now made of brass.

The Admiralty, and consequently all foreign governments, require the brass tubes to be tinned when the condenser is of iron, and some of the large steamship companies also continue this practice, but, as a rule, in the mercantile marine, the tubes are untinned. The tinning adds 1½d. per pound extra to the cost of the tubes, and amounts to an increase of 16 per cent.; it is not necessary, but is only an additional safeguard against the formation of copper salts on the one hand, and to the corrosive action of sea-water on the other. Brass tubes, untinned, after twelve years' constant use, have been found on being cleaned, to be nearly as good as when new; on the other hand, brass tubes have pitted badly, and in places been perforated in a few months from causes practically unknown. It is conjectured that galvanic action has set up from iron filings, carried on to the tube surface by the steam or circulating water, causing a separation of the copper and zinc and the dissolving away of the latter. Microscopic examination will probably throw much light on this theory, and eventually clear up what is now somewhat of a mystery. As the condensers of warships are now always made of brass, the tubes are untinned, but the Admiralty require the addition of 1 per cent. of tin to the mixture of copper and zinc, and this composition does not pit or corrode.

As will be shown later on, the loss from blowing off the boilers to prevent dangerous incrustation when fed from the hot-well of a jet-condenser, amounted to as much as 25 per cent., and seldom less in general practice than 12 per cent. This loss is almost wholly avoided by the use of a surface-condenser, and an additional saving of no mean importance is effected in avoiding the necessity of so often stopping to scale and clean out the boilers, as was the case when jet-condensers were used. *The net saving of fuel* by the use of a surface-condenser averages 15 per cent.; and in the hands of a careful engineer, the economy may be extended to even 20 per cent.

Just as 30 or 35 lbs. pressure is the limit for which a box boiler may be safely and conveniently constructed, so that is the limit at which a boiler can safely supply steam to a jet-condensing engine using sea-water. When sea-water is raised to a temperature of 280° Fahr., which corresponds to a pressure of 50 lbs. absolute, or 35 lbs. above the atmosphere, what are called its *insoluble salts* are wholly precipitated, and form a hard scale on the hot surfaces. The principal insoluble salt in sea-water (*v.* Chapter xxv.) is sulphate of lime; it is called insoluble, because it does not dissolve in water under ordinary circumstances, and consequently when deposited on the surface of the tubes, &c., will not redissolve and wash off again. The carbonate of lime, and the salts of soda and magnesia, are comparatively harmless, for although the former is precipitated, it is only in a soft muddy state, and mixed with the brine products of the latter can be blown off, and easily removed from the boiler when in port. It is for this reason that a surface-condenser is an imperative necessity for engines using steam above 35 lbs. pressure, or 50 lbs. absolute.

For the same reason it is advisable to fit a surface-condenser to steamers running in muddy rivers, or on lakes whose water is dense, as otherwise the boiler soon fills with deposit, which, unless removed and the boiler thoroughly cleaned, will cause serious damage, and be a source of danger as well as a constant cause of priming.

It is seen, then, that by the use of a surface-condenser steam of higher pressure than 50 lbs. absolute may be employed; a considerable saving of fuel and time is effected; and there is considerably less risk of burning and otherwise damaging the boiler. A better vacuum is also obtained in it, as a rule, than was possible with the jet-condenser. On the other hand, a surface-condenser is heavier, more costly, and occupies more space than the jet-condenser; a second pump for the cooling water is necessary, and although the air-pump need not be so large as for a jet-condenser, it was often made so in case the jet was used or the tubes leak. The wear and tear and the store account are increased somewhat when there is a surface-condenser, and more care and responsibility are laid on the engineers; but these are mere trifles compared with the benefits, and are only mentioned now as examples of the reasons used by the opponents of Mr. Hall's system in the bitter controversy that raged in the forties of last century, and prevented the general adoption of it. For sea-going vessels the surface-condenser is indispensable and always fitted, but in very fast ships on short voyages, where weight of machinery was of more consequence than economy of fuel, and where the weight of coal carried is small, the jet-condenser was often retained. Now, however, even these ships are fitted with surface-condensers.

Cooling Surface.—Professor Rankine suggested the following as a means of ascertaining the amount of cooling surface:—Let t denote the temperature of a film of liquid at one side of a metal plate, s_1 the extent of cooling surface; let heat be communicated to the liquid at a temperature t by some such process as the condensation of steam, and let that be abstracted by the flow of a current of air, water, or other fluid, in contact with the metal plate; the weight of fluid which flows past per second being W, its specific heat c', its initial temperature T_1, being lower than t, but higher than T_2. Then in all the equations $t - T_1$ is to be substituted for $T_1 - t$, and $t - T_2$ for $T_2 - t$ in the equation $\frac{S}{c\,W} = a\left\{\frac{1}{T_2 - t} - \frac{1}{T_1 - t}\right\}$; but he also adds that there are not sufficient data to obtain the value of the constants.

Peclet found that with cooling water of an initial temperature of 68° to 77°, one square foot of copper-plate condensed 21·5 lbs. of steam per hour, while Joule stated that 100 lbs. per hour can be condensed. In practice, with the triple-compound engine, brass condenser tubes, 18 B.W.G. thick, 15 lbs. of steam per square foot per hour, with the cooling water at an initial temperature of 60°, is considered very fair work when the temperature of the feed-water is to be maintained at 120°. As much as 25 lbs. per square foot is condensed in "destroyers," and it is not unlikely that this quantity could be kept up so long as the tubes are clean and the circulation rapid. It follows, therefore, that with engines exhausting at a lower temperature a still greater weight per sq. ft. would be converted to water. Professor Weighton with the "Contraflo" condenser got 33 lbs. of water per sq. ft. from a triple engine.

In the early days of surface-condensation, it was customary to provide as much cooling surface in the condenser as there was heating surface in the boiler, and a general fear of consequences caused engineers to give a too liberal allowance of surface long after experience had shown that much less would do. The surface in a condenser is much more efficient in abstracting heat than can be that of a boiler. The tubes of a boiler are more than double the thickness of the condenser tubes, and the plates more than eight times; and while the condenser will keep clean with care for a very long time, the boiler gets dirty in a few hours by soot and sooty scale on the one side, and salt incrustation and rust on the other. For these reasons the surface in the condenser may be less than a half of that in the boiler, and under some circumstances even considerably less than this.

In general practice the following suffices when the temperature of sea-water is about 60°, and allows for the surface getting dirty:—

Terminal pressure,	20	lbs. absolute,	1·57	square feet	per I.H.P.
"	15	"	1·42	"	"
"	12½	"	1·28	"	"
"	10	"	1·14	"	"
"	8	"	1·00	"	"
"	6	"	0·80	"	"

For ships whose station is in the tropics the allowance should be increased by 10 per cent. If a ship is constantly employed in cold climates, 20 per cent. less suffices. With cooling water not exceeding 70°, the surface need only be one square foot for every 20 lbs. of water to be condensed. Thus, with a triple-compound engine using 13 lbs. of steam per I.H.P., 0·65 square foot of surface per I.H.P. would suffice. One square foot per I.H.P. on trial trip is now deemed ample for ships going to any part of the world where the sea-water does not exceed 85° F.

Condenser Tubes.—They are, as a rule, made of brass, solid drawn, and tested both by hydraulic pressure and steam; the latter test is a very useful one, as faults which escape detection under water pressure are often found out by steam; these faults are due generally to minute particles of flux or slag in the original ingot, and sometimes the faults are in the form of cracks done in the process of drawing. It is by no means an uncommon thing to find a few tubes in a new condenser leaking through minute holes of various shapes; these holes soon become enlarged if the tube is not at once stopped or withdrawn. These faults are not confined to the tubes of a few makers, but may be found in those of all makers at some time or other. Tinning is a preventive, as the defective places are in the process covered or filled with that metal.

Condenser tubes are usually made of a composition of 68 per cent. of best selected copper, and 32 per cent. of best Silesian spelter. The Admiralty, however, always specify the tubes to be made of 70 per cent. of best selected copper, and to have 1 per cent. of tin in the composition, and test the tubes to a pressure of 300 lbs. per square inch. To prove that the tubes are of the required metal, a few pounds of them are melted in a closed crucible, and sufficient spelter added to bring the mixture to contain 62 per cent. of copper. The metal is then cast into an ingot, which when cold is rolled into a sheet, strips are cut from it and tested, and if satisfactory should have an ultimate tensile strength of 24 tons per square inch. The diameter of the condenser-tubes varies from ½ inch in small Naval condensers, when they are very short, to 1 inch in large mercantile condensers and long tubes. In the mercantile marine the tubes are, as a rule, ¾ inch diameter externally, and 18 L.S.G. thick (0·049 inch); and 16 L.S.G. (0·065), under some exceptional circumstances. In H.M. Navy, the tubes were also, as a rule, ¾ inch diameter, and 18 to 19 L.S.G. thick, tinned on both sides; when the condenser is made of brass the Admiralty do not require the tubes to be tinned. On account of the economy of space and weight that is effected with small tubes all Naval engines are now fitted with condensers having tubes ⅝ inch diameter. The smaller the tubes, the larger is the surface which can be got in a certain space. Since larger tubes are of necessity somewhat thicker than the smaller ones, a square foot of surface costs more when they are adopted, and is not so efficient. Patent tubes made from sheet brass 22 B.W.G. thick, and joined at the seams like a tinsmith's joint and soft soldered, have been tried. The advantage claimed for them is the uniformity of thinness, whereby as little as 22 B.W.G. is sufficient thickness, while it would not be safe to use drawn tubes of that size.

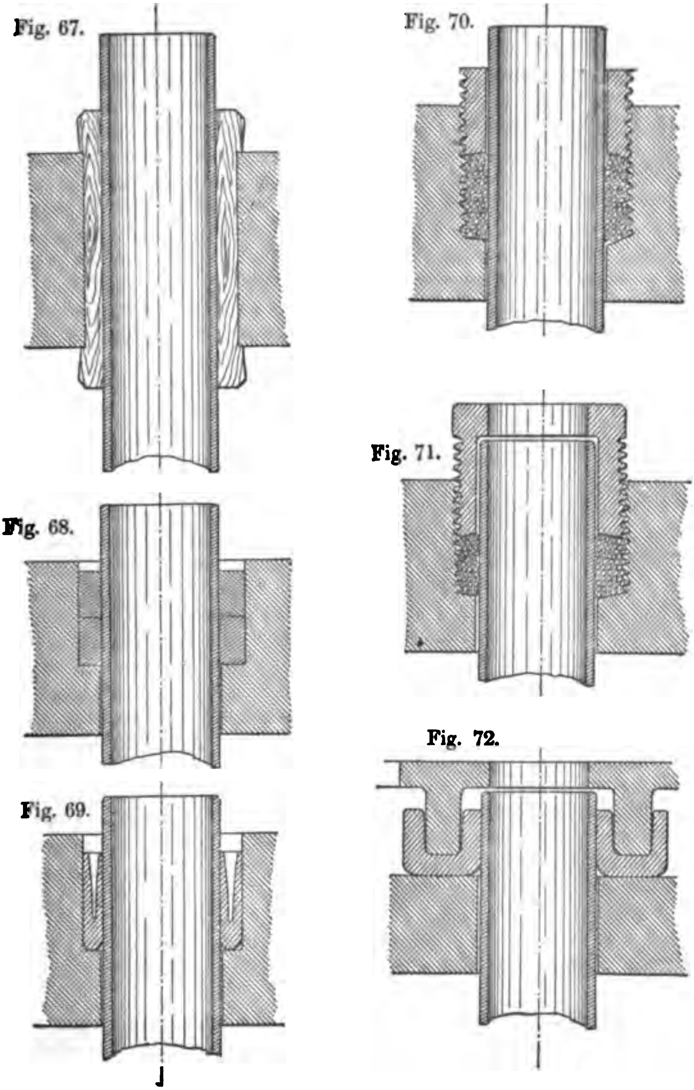
The length of the tubes depends on the arrangement of the condenser, but when they are not held tightly in the plates, but only packed, their unsupported length should not exceed 100 diameters; when held with tight-fitting ferrules it may be 120 diameters.

Tube-plates are now always made of brass, either cast or rolled into plates of suitable size; the latter is preferable, as the rolled brass is very tough and close grained, and as strong as wrought iron. Formerly it was no uncommon thing to make the tube-plates of cast iron from 1¾ to 2¼ inches thick, and while some were converted into a substance resembling plumbago after two or three years' work, others have been found sound and good after twelve years' continuous work.

Rolled brass tube-plates should be from 1·3 to 1·5 times the diameter of tubes in thickness, depending on the method of packing. When the packings go completely through the plates the latter, but when only partly through, the former is sufficient. Hence, for ¾-inch tubes the plates are usually ⅞ to 1 inch thick with glands and tape-packings, and 1 to 1¼ inches thick with wooden ferrules. In the Navy the tube-plates are generally 1 to ⅞ inch thick, the tubes being ⅝ inch diameter and 19 L.S.G. thick, but in the "Destroyers" the plates are only ¾ inch thick; in their case, however, the plates are of small diameter and stayed in the middle; and it may be added that leakage of tubes is no uncommon occurrence in these ships, so that this is as thin as they can safely be employed.

The tube-plates should be secured to their seatings by brass studs and nuts, or brass screw-bolts; in fact, there must be no wrought iron of any kind on the sea-water side of a condenser. When the tube-plates are of large area it is advisable to stay them by brass rods, to prevent them from collapsing.

Tube Packings.—All attempts to drift or expand the tubes tightly into holes in a brass plate fail, owing to the softness of both plates and tubes; and if it could be done it would be found impossible to draw the tubes for examination and cleaning without damage. Fig. 67 shows a very simple plan, and one that has proved effective under all circumstances. The ferrule is made of soft wood, such as pine or lime tree, very dry and well seasoned; they are made nearly an eighth of an inch larger in diameter than the hole into which they have to fit, and are a good fit on the tube. Before fitting them into place they are squeezed through a die in a press until they can be easily driven into their holes; soon after being fitted into place they absorb moisture and expand circumferentially at each end, and become exceedingly tight on the tube and



Figs. 67-72.—Condenser Tube Packings.

in the hole. After twelve years' service they are found quite sound. It is urged against them that they are apt to shrink and drop out when the condenser is not in use, but this is not the case, as the swelled projecting ends form collars to prevent this, and they do not shrink so much as is generally supposed, unless by unusual heat. This is one of the cheapest forms of tube-packing, and although not used now in H.M. Navy is often employed in the mercantile marine of this and other countries, and does very well when the tubes are not very long; these ferrules also specially suited the cast-iron tube-plate very well. At present they are seldom used in new engines.

The next simple method (fig. 68) consists of two square section india-rubber washers, fitting tightly on the tube ends, and driven into a recess in the tube-plate. This method is a fairly good one when the water circulates through the tubes, for then the washers are pressed into their place, and are not exposed to the action of grease. Fig. 69 shows an extension of this method; the washers are made like the cup leather washers for hydraulic machinery, and the water pressure forces the india-rubber tightly against both tube and recess. This plan admits of the tubes being quickly and easily withdrawn, but will not do when the tubes are placed vertically, or when the water is outside the tubes.

Sometimes a sheet of india-rubber (fig. 72), having small holes corresponding to the tubes, was laid on the tube-plates, and forced over the tube ends, and secured by a brass cover plate, having a recess into which each tube end fits, and a hole through the bottom of the recess corresponding to the bore of the tube. This was by no means an economical plan, and it has also the serious objection that if one tube leaks at the joint the whole have to be disturbed in the attempt to stop the leak. The plan adopted in H.M. Navy, and very generally in the mercantile marine, is that shown in fig. 70. Each tube end passes through a stuffing-box fitted with a screwed gland, and kept tight by a tape washer, or some soft cord as packing. This method is the most expensive of all, as the tube-plate is twice drilled and all the holes tapped; the screwed ferrules are expensive, and the labour of packing considerable compared with that of the other methods; also many of the glands are destroyed when the condenser tubes are removed for cleaning. This plan, however, admits of the water being on either side of the tubes; the packing is not affected by heat, and the condenser may remain unused for a very long time, and be quite tight at the end of it; for these reasons it was chosen by the Admiralty. It is likewise the plan used by Hall in his early surface-condensers, and on the whole the most satisfactory one. Modern methods of drilling and tapping the holes, making the ferrules, &c., have very much reduced the cost, and so practically removed the only objection. Fig. 71 shows a modification of gland to suit vertical tubes; the gland has an inside rim, which prevents the tube from slipping. It is now always used in condensers with horizontal tubes, and is especially necessary when they are long.

Steam Side and Water Side of Tubes.—This was somewhat of a vexed question, and one on which there is much to be said on both sides. The naval practice was to circulate the water *outside* the tubes, so that the condenser shell may be kept cool and prevented from making the engine-room hotter than can be helped. The almost universal practice of the merchant service and Navy now is to circulate the water *through* the tubes. Independently of the particular reason for the choice of the Admiralty, the balance of argument is in favour of circulating the water through the tubes;

for when this is the case there is (i.) a larger surface of metal exposed to the hot steam ; (ii.) the grease deposited on the tubes is easily removed by working a trifle warm, and using a solution of caustic soda or potash, and if this does not remove it, the deposit being soft does not prevent the tubes from being easily drawn, as is the case when scale from salt water is deposited on their exterior surface ; (iii.) the scale from sea-water, which must be removed mechanically, can be so done without removing the tubes ; (iv.) a more extended and complete circulation of the cooling water is possible, and that without risk of air accumulation, and without special and expensive diaphragms, &c. ; (v.) the condenser is more easily designed, and fits into the general arrangement of most engines, and is smaller, inasmuch as there is no need of an expansion chamber in front of the tubes, as is the case when steam passes through them ; (vi.) when it is necessary to examine the packings, or to plug a defective tube, only a water joint is broken ; (vii.) india-rubber packings cannot be used when the steam passes through the tubes, as they are destroyed by the fatty matter deposited by the steam, and the sulphurous products of their decomposition attack and destroy the tube ends, &c. ; and (viii.) the thin tubes are stronger to resist internal than external pressures.

On the other hand, when the steam passes through the tubes the bulk of the fatty matter is deposited on the front tube-plate, and prevented from coating the tubes ; the large flat sides of the condenser are subject to the very slight pressure due to the head of water, and so may be made much lighter ; and there is less hot surface exposed in the engine-room. As little or no oil is now used for direct internal lubrication, and such as gets in with the piston and valve rods is small, and a mineral oil, the argument as regards grease deposit is considerably modified from what it was when animal and vegetable oils were freely used, and even tallow on occasion, in both cylinders and boilers.

Spacing of Tubes, &c.—The holes for ferrules, glands, or india-rubber are usually $\frac{1}{4}$ inch larger in diameter than the tubes ; but when absolutely necessary the wood ferrules may be only $\frac{3}{32}$ inch thick.

TABLE XIX.

Pitch of Tubes.	No. in a square foot.	Pitch of Tubes.	No. in a square foot.	Pitch of Tubes.	No. in a square foot.
Inches.		Inches.		Inches.	
$\frac{31}{32}$	184	...	...	...	...
1	172	$1\frac{5}{32}$	128	$1\frac{1}{4}$	110
$1\frac{1}{32}$	161	...	...	...	...
$1\frac{1}{16}$	152	$1\frac{3}{16}$	122	$1\frac{9}{32}$	105
$1\frac{3}{32}$	144	...	...	...	...
$1\frac{1}{8}$	136	$1\frac{7}{32}$	116	$1\frac{5}{16}$	99

The pitch of tubes when packed with wood ferrules is usually $\frac{1}{4}$ inch more than the diameter of the ferrule hole. For example, the tubes being $\frac{3}{4}$ inch external diameter, the ferrule hole will be 1 inch and the pitch of the tubes $1\frac{1}{4}$ inch. When the india-rubber washers or screwed glands are used, the pitch of the tubes is $\frac{1}{8}$ inch to $\frac{3}{16}$ inch more than their diameter, so that the usual pitch for $\frac{3}{4}$-inch tubes under these circumstances is $1\frac{5}{32}$ inch. In the Navy, with tubes $\frac{5}{8}$ inch diameter and the tube-plates $\frac{7}{8}$ inch to 1 inch thick, the pitch of holes is $\frac{31}{32}$ inch, and in "Destroyers" only $\frac{15}{16}$ inch. In the mercantile marine, with tubes $\frac{3}{4}$ inch diameter and the tube-plates 1 inch thick, the pitch is $1\frac{1}{4}$ inch; for $\frac{5}{8}$-inch tubes, $1\frac{5}{16}$ inch; and for 1-inch tubes, $1\frac{7}{16}$ inch. The tubes are generally arranged zigzag, and the number which may be fitted into a square foot of plate is as in Table xix.

The Condenser.—The surface-condenser is generally in the form of a cylinder or a rectangular parallelepiped, and sometimes a flattened cylinder; the first and last forms are the best suited when weight is a great consideration, and the second the most convenient when space is of first importance; the cylindrical form is by far the cheapest, as the patterns are very simple—the body, when not made of iron, copper, or brass sheets, being *struck out;* the covers, tube-plates, and corresponding flanges can all be faced in a lathe; this form also is by far the lightest, for the two reasons, that the circular plate is the form giving the minimum perimeter for a given area, and consequently a minimum barrel, and that the cylindrical form for either internal or external pressure is the strongest, so that the thickness of metal will be the minimum.

The waterways, or chambers at each end of the condenser, are sometimes cast with it, and sometimes cast separately; in the latter case there is an additional joint, but this is mitigated by its also forming the tube-plate joint; in the former case there is only one joint less at each end through which air can leak, but the plates are more troublesome to fit, and, except in the case of the cylindrical form, the tube-plate flange is difficult to face.

Great care should be taken in designing a condenser that free outlet is given to any air that may collect near the tubes, and all pockets, where dead water could lie, should be avoided, as the vacuum is very often spoiled by a few of the tubes getting hot and leaking from the above causes.

The hottest water should meet the hottest steam, so that the inlet should be at that part where the condensed vapour leaves the condenser. Some engineers object to this on the ground that the condensed water is unnecessarily cooled, and, to avoid this, cause the coldest water to meet the hottest steam, and, as the circulating water gradually becomes warmed, the condensed water is not cooled so much as in the other case. Now, this practice is not in accordance with theory, and if the matter were looked into a little closer it would be found that although the observations of practice were correct the inferences were wrong. The pressure in the condenser will depend on the temperature of the vapour, and if the temperature of the condensed water were higher than that due to the pressure, vapour would be immediately formed, and the vacuum fall until the pressure became that due to the temperature of the condensed water; of course, it is possible that the condensed water may have a temperature considerably below that of the vapour, but this would only occur in a badly designed or badly worked condenser; in other words, the condensed water is allowed to remain in contact with the cold tubes longer than is necessary, owing to imperfect drainage, and its coolness is not directly due to the system of circulation. Now, if the water is admitted to the hottest part of the condenser first, the absorption of heat, owing to the wide difference of temperatures, will be considerably more than when the cooling water has become warmed and the vapour cooled, so that their difference of temperature is not so wide; when this is the case, to reduce the vapour to the temperature due to the pressure required, either a larger surface or more cooling water is necessary. Now, if the coldest water meet the coldest vapour, and the condensed water is well drained as soon as

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formed, the difference between the temperature of the condensing water and the vapour will not vary very much from entrance to outlet, and less surface and less condensing water will be required; but whichever plan is adopted, if the condensed water is allowed to remain long in contact with the tubes, it will be unnecessarily cooled, and give cold feed water. There is, however, little fear of such an action in a condenser with small surface and as now designed.

Quantity of Cooling Water.—The necessary amount of circulating water may be calculated in the same way as that for injection water (see p. 225), on the principle that the exhaust steam has a certain quantity of heat which is to be expended in raising a mass of sea-water of a certain temperature to about 100°. The quantity of sea-water will of course depend on its initial temperature, which in actual practice may vary from 40° in the winter of temperate zones to 80° of the West Indies and other subtropical seas. In the latter case a pound of water requires only 20 thermal units to raise it to 100°, while 60 are required in the former. From this it is seen that the quantity of circulating water required in the tropics is three times that of the North Atlantic in the spring of the year.

As before, let T_1 be temperature of the steam on entering the condenser, and L the latent heat; T_0 the temperature of the circulating water, and Q its quantity; T_2 the temperature of the water on leaving the condenser, and T_3 the temperature of the feed.

The heat to be absorbed by the cooling water is now $(T_1 + L) - T_3$; and this amount of heat must be equal to $Q\ (T_2 - T_0)$. Hence,

$$Q = (T_1 + L) - T_3 \div T_2 - T_0.$$

Or

$$Q = \frac{1114 + 0{\cdot}3\,T_1 - T_3}{T_2 - T_0}.$$

Example.—To find the amount of circulating water required by an engine whose steam exhausts at 8 lbs. pressure absolute, the temperature of the sea being 60°, and (2) the amount required when the temperature of the sea is 75°. The temperature of the feed to be 120°, and that of the water at the discharge 100°. The temperature corresponding to 8 lbs. is 183°.

(1)

$$Q = \frac{1114 + 0{\cdot}3 \times 183 - 120}{100 - 60} = 26{\cdot}22.$$

That is, the water required is 26·22 times the weight of steam.

With the jet condenser, the quantity was only 17·48 times (*v.* p. 226).

(2) When the sea is at 75°

$$Q = \frac{1114 + 0{\cdot}3 \times 183 - 120}{100 - 75} = 41{\cdot}95 \text{ times.}$$

It is usual to provide pumping power sufficient to supply 30 times the weight of steam for general traders, and as much as 40 times for ships stationed in subtropical seas. As will be shown in another chapter, if the circulating pump is double-acting, its capacity may be $\frac{1}{53}$ in the former, and $\frac{1}{42}$ in the latter case of the capacity of the low-pressure cylinder.

Passage of Circulating Water.—The water must be caused to pass over a sufficient amount of surface to become duly heated, if the minimum quantity is to be used. In practice it should travel at least 20 feet lineally through the tubes before leaving the condenser; if this cannot be arranged, then it must remain longer in contact with the surface. Hence, in small condensers, where the steam is outside the tubes, the water circulates only twice through them at a slow pace; in larger condensers it may circulate twice through long tubes, or three or four times through shorter tubes at a higher velocity, due to the larger quantity of water. To obtain the best results the velocity of flow should be 110 feet per minute when the sea-water is at 60°, and 170 feet when at 75°. When the water circulated outside the tubes, it was necessary to fit baffle-plates, which should divert the water and prevent it from taking the shortest course from the inlet to the outlet, and also to prevent the hot water from accumulating at the top part and the cold water from remaining in the bottom. Some skill and ingenuity were often required to fully overcome such difficulties. Condensers are, however, seldom or never made now with the water outside the tubes.

Where reciprocating pumps are employed, the circulating water should be admitted to the condenser direct from the sea, and *pumped from it* through an outlet above the level of the top row of tubes; the pressure in the condenser does not then exceed that due to the head of water, and little or no shock is given to it by the varying velocity of the circulating pump, as is the case when the water is forced through. When a centrifugal pump is used, it should deliver into the condenser direct from the sea.

Size of Inlet and Discharge Pipes should be such that the flow of water through them when the engines are working at full speed does not exceed 500 feet per minute. In temperate zones 7 lbs. of cooling water per I.H.P. is sufficient for a triple-compound engine, and 6 lbs. will do for a quadruple engine or a triple working at a fairly high rate of expansion. Hence

$$\text{Area section inlet and discharge} = \frac{\text{I.H.P.}}{\text{C}} \text{ sq. ins.}$$

Where C = 31 triple and quadruple high expansion,
= 30 „ „ moderate „
= 25 „ „ tropics.

Extra Supply Cock.—To provide for the water wasted in blowing off, priming, leakage, &c., it is usual to fit a small cock, through which some of the circulating water may be passed to the steam side of the tubes. The pipe for this should be about one-third the diameter of the main feed pipe, the velocity of flow being nearly nine times that usually provided for in feed pipes. Now that evaporators are in general use, so that the waste can be made up with fresh water, the necessity for such a fitting scarcely exists, but as an emergency provision it may remain.

Impermeator.—A cup with cock attached should be fitted to the condenser close to the exhaust entry, by means of which caustic soda may be injected to the condenser, and spread over the tube surface to dissolve off grease.

Man-holes and Mud-holes.—A man-hole is necessary for the purpose of admitting men to clean, repair, or tube the condenser, and smaller holes should be provided through which mud, grease, scale, &c., may be scraped out. Peep-holes are sometimes formed in the doors, especially if they are large and heavy, through which the tube ends may be seen and examined. These latter are useful when steam is condensed inside the tubes, to admit the nozzle of a steam jet to wash away grease, &c.

Drain Cocks should be fitted so that the condenser may be thoroughly drained when not in use.

Testing.—The Admiralty require condensers to be tested to 30 lbs. per square inch before being placed in the ship, and many steamship companies require the same test, while others are content to test with lower pressures. To provide for such strains, the flat surfaces must be stiffened in the same way as laid down for cylinders, and, when necessary, tied together by stay bars, &c.

Cementing.—It is a good plan to cover the inside of iron condensers, where there is much wash of condensed water, with a good coating of Portland cement, and under the air-pump and in the pump passages it should be at least half an inch thick.

Evaporators.—Since the use of steam of 150 lbs. and upwards the extra supply from the sea has been avoided as much as possible, fresh water being carried in tanks or double bottoms; now the employment of "evaporators" is doing away with the necessity for this and providing a long-felt want, and permitting of the use of water-tube boilers and modified forms of ordinary marine types.

CHAPTER XII.

PUMPS.

Air Pump.—The function of this pump in all condensers is to abstract the water condensed, and the air which was originally contained in the water when it entered the boiler; and in the case of jet-condensers, it pumps out in addition the water of condensation and the air which it contained.

It follows then that the size of the air-pump must be calculated from these conditions, and allowance made for the efficiency of the pump; or, what is the same thing, the result thus found must be multiplied by the ratio between what the pump should do theoretically, supposing its action perfect, and what it does actually in practice.

Ordinary sea water contains, mechanically mixed with it, one-twentieth of its volume of air, when under the atmospheric pressure. Now, suppose the pressure in the condenser to be 2 pounds, and the atmospheric pressure 15 pounds, neglecting the effect of temperature, the air on entering the condenser will be expanded to $\frac{15}{2}$ times its original volume; so that a cubic foot of sea-water, when it has entered the condenser, is represented by a cubic foot of water, and $\frac{15}{40}$ of a cubic foot of air.

Now let q be the volume of water condensed per minute, and Q the volume of sea-water required to condense it; and let T_2 be the temperature of the condenser, and T_1 that of the sea-water:—

Then $(q + Q)$ will be the volume of water to be pumped from the condenser per minute, and

$$\frac{15}{40}(q + Q) \times \frac{T_2 + 461^\circ}{T_1 + 461^\circ} \text{ the quantity of air.}^{*}$$

---

[*] Absolute zero point, or point of no heat, is 461° below the zero of Fahrenheit's thermometer.

If the temperature of the condenser be taken at 120°, and that of sea water at 60°, the quantity of air will then be $\cdot 418\ (q + Q)$, so that the total volume to be abstracted will be

$$(q + Q) + \cdot 418\ (q + Q) = 1\cdot 418\ (q + Q).$$

Now, if the average quantity of injection water be taken at 26 times that condensed, $q + Q$ will equal $27\ q$.

Therefore, volume to be pumped from the condenser per minute

$$= 38\ q.$$

Example.—To find the theoretical capacity of a single-acting pump for an engine using 3 cubic feet of water per minute, and the number of strokes being 40.

Volume to be pumped out = 38 × 3, or 114 cubic feet.

Therefore, the capacity of the pump $= \frac{114}{40}$, or 2·85 cubic feet.

If the stroke of the pump be taken at 1·5 feet.

The area of bucket $= \frac{2\cdot 85}{1\cdot 5}$, or 1·9 square feet.

Example.—To find the theoretical capacity of a double-acting pump for an engine using 10 cubic feet of water per minute, the number of revolutions being 90, and the stroke of the pump 2·5 feet.

Volume to be pumped out = 38 × 10, or 380 cubic feet.

Area of bucket $= \frac{380}{90 \times 2 \times 2\cdot 5}$, or ·844 square foot.

Of course were these examples worked strictly, it would be necessary to find the relation between Q and q in each case, instead of assuming it at 26 as has been done.

The foregoing calculations, &c., are for jet-condensers, and based on the supposition that the air-pump also abstracts the condensing water; now in a surface-condenser, not only is the air-pump relieved of this latter duty, but, since the feed-water is condensed steam, and has not had time to absorb air, it would seem that its only function is to draw off the condensed water, and should therefore be no larger than a feed pump, as no doubt it might be, did the engine work perfectly; but since ordinary water has to be occasionally admitted to make up for waste, and as slight leakage at the glands and joints very frequently exists, it would be necessary to make the air-pump only about half the size that would be requisite for jet condensation; but when a surface-condenser was arranged so as to be worked as a jet-condenser, the air-pump had to be large enough to do the work necessitated by this, unless the circulating pump was fitted so as to be worked as an air-pump. It is seldom that a surface-condenser is now so arranged as to admit of jet condensation. Too great care cannot be expended on the design of the air-pump for a surface-condenser, as the success of many an engine has been marred by a bad vacuum; and doubtless an inch or two of vacuum tells wonderfully on an engine, especially on a compound engine, where the vacuum can almost be told by the speed of the engine. Since, however, the surface-condenser was only required to work as a jet-condenser, when so fitted, in cases of emergency, which seldom happen, and when such a case does occur it is not of importance that the engines shall work at the highest efficiency or maximum speed, it seems better so to design the air-pump as to best suit surface condensation, rather than to make it of the larger size, and sacrifice a large amount of work in driving it during the long period, when jet condensation is not necessary. In short, the pump should be of such a size as to give its maximum efficiency during the longest time, and if for a surface-condensing engine, it should be designed for that particular service.

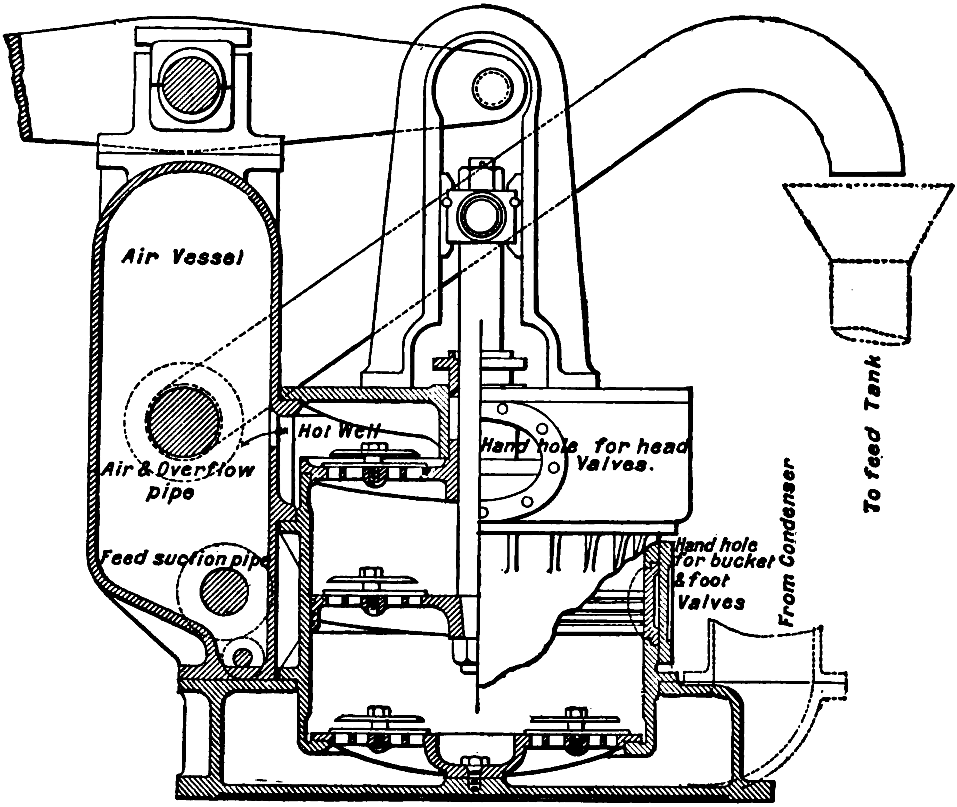

Fig. 73.—Air-pump of Ordinary Type.

Such reasoning did not rigidly apply to the machinery of warships, as a mishap in action might necessitate the use of the jet injection, and at such a time there must be no diminution of speed and efficiency; for this reason it was customary for the Admiralty to stipulate that the air-pump should be of sufficient size to produce a good vacuum when jet condensation was resorted to. But the Admiralty no longer provide for working with jet injection.

In the mercantile marine, especially in mail and passenger steamers, it was customary to effect a compromise by making the pump larger than absolutely necessary for surface condensation, but less than is usual for that purpose.

The Single-acting Vertical Air-pump (fig. 73), having valves in the bucket as well as foot and delivery valves, is by far the most efficient, and, when possible, was generally chosen. If the rod of such a pump is enlarged or the bucket has a trunk to surround the rod, which is attached to a joint in its centre, it is to a certain extent double-acting, since on the upstroke it fills the chamber in which it works, on the downstroke it displaces, and consequently discharges a volume equal to the volume of the trunk or rod, and on the upstroke discharges the remainder. If the sectional area of the rod or trunk is half that of the bucket, the discharges are equal, and the pump is virtually a double-acting one.

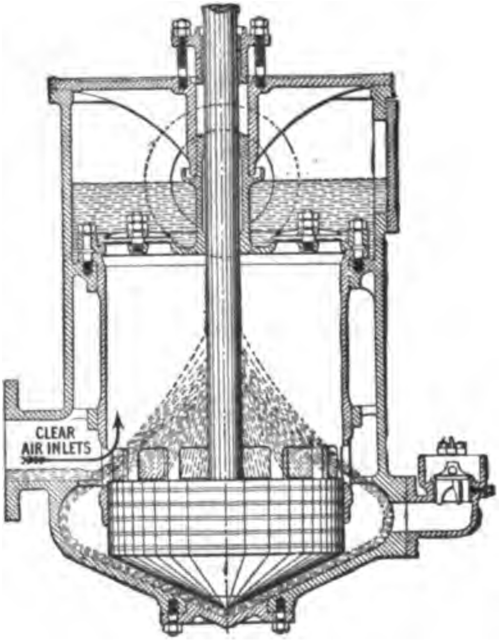

Fig. 74.—Edwards' Air-pump. Bucket at Bottom of Stroke.

Edwards' Air-pump is an ingenious form of single-acting vertical pump, with valves only at the top to check the discharged water and air from returning to the pump on the down stroke. Fig. 74 shows the general design and action of the bucket or piston of this pump whereby the water and air are caused to enter the exhausted space between the head-valves and the bucket. These pumps are very simple, have the minimum number of valves, which are easily examined and replaced, work with the least possible resistance and wear and tear, and produce a good vacuum.

The Double-acting Air-pump.—It is not always convenient to have a vertical pump in the horizontal engine, and consequently a horizontal pump was generally employed, and this was almost of necessity double-acting. The bucket in this case works air-tight in a smooth barrel placed beneath the condenser, and has a set of foot and delivery valves for each end. Sometimes, in lieu of a barrel and bucket, a plunger is fitted, passing through a stuffing-box in the diaphragm-plate dividing the condenser-bottom. This latter arrangement generally admits of more room for, and a better disposition of, the valves. Some engineers adopted this form of the horizontal pump for vertical engines, and worked it by means of an eccentric formed on one of the crank-arms.

The Efficiency of Air-pumps.—The efficiency of the single-acting vertical pump is due to the certainty of its action in taking the water, &c., through the bucket-valves, and to the valves from their position so readily closing when required; there is also time for the water to drain into the bottom of the pump on its upstroke, and collect there ready for the bucket when it descends. The flow is always in one direction, so that the velocity of flow is not checked by diversion. The water always lies on the valves so as to render them air-tight, and there is very little clearance space, as a rule, between the foot and bucket valves, and between the bucket and head valves, and what there is contains water.

The want of efficiency of the double-acting horizontal pump is caused by the reverse of some of the above conditions, especially by the failure of the valves in closing, and to the large space between the foot and delivery valves, also by leakage at the gland of the rod, and past the bucket, which is only lubricated by the water on the bottom, and in no small degree by the ever-changing direction of flow. The latter defect is proved by stopping one end of the pump, when it is often found (especially in the case of badly designed pumps, &c.) that the vacuum is not very materially altered. The foot valves are sometimes kept covered with water by allowing the water to pass back again through a pipe from the hot-well to the pump-chamber.

Size of Air-pump.—The capacity of the air-pump should be calculated from consideration of the conditions under which it is to work, and by the rules given in this and the preceding chapter, and suited to practice by an allowance made for the efficiency of the pump. If the pump is single-acting and well-designed, and is working under favourable conditions, its efficiency may be taken at 0·6; and if the reverse of this 0·4; generally its efficiency is about 0·5, so that the size in practice should be double that given by theoretical calculation. The efficiency of the double-acting pump varies from 0·5 to 0·3, and generally is not more than 0·35; the size for good working should be nearly three times that given by theoretical calculations.

Hence, when the temperature of the sea is 60°, and that of the (jet) condenser is 120°, Q being the *volume* of the cooling water, and q the *volume* of the condensed water in cubic feet, and n the number of *strokes* per minute.

$$\text{The volume of the single-acting pump} = 2{\cdot}74\left(\frac{Q+q}{n}\right).$$

$$\text{The volume of the double-acting pump} = 4\left(\frac{Q+q}{n}\right).$$

In actual practice it is usual to make the air-pump proportional to the capacity of the cylinder or cylinders exhausting into the condenser. The old rule for the air-pumps of paddle-wheel engines supplied with steam of 20 lbs. pressure was to make the diameter of each pump half that of each cylinder, and the stroke half the stroke of the piston; in other words, the capacity of the pump was one-eighth that of the cylinder. For example, a paddle-wheel engine having two cylinders 80 inches diameter and 72 inches stroke, would have two single-acting air-pumps, 40 inches diameter and 36 inches stroke.

The following table gives the ratio of capacity of cylinder or cylinders to that of the air-pump; in the case of the compound engines, the low-pressure cylinder capacity only is taken. A small air-pump will keep up a good vacuum in a surface-condenser, and require less power to drive than a larger one, but it takes longer to produce a good vacuum. On the whole, a small pump is best, and now that good metallic packings are fitted to the principal glands, a satisfactory vacuum can be maintained with it.

TABLE XX.

Description of Pump.	Description of Engine.	Ratio.
Single-acting vertical,	Jet-condensing, expansion, 1½ to 2,	6 to 8
" "	Surface " " "	10 to 12
" "	Jet " " 3 to 5,	10 to 12
" "	Surface " " "	14 to 16
" "	" " compound, 6 to 10,	18 to 21
" "	" " triple and quadrpl., 10 to 16,	21 to 28
Double-acting horizontal,	Jet " expansion, 3 to 5,	16 to 19
" "	Surface " " "	19 to 24
" "	" " compound, 6 to 10,	24 to 28
" "	" " triple-ex., 10 to 16,	28 to 30

The stroke of the vertical air-pump is usually from one-third to one-half that of the engine, but it should be such that the velocity does not exceed 300 feet per minute, and for continuous running 275 feet.

The vacuum in the condenser is liable to be spoiled by leakage at the expansion joint of the exhaust pipe. It is usual to provide for the expansion of this pipe, which, in the Navy, is of brass or copper, by fitting a stuffing-box, gland, &c., on the condenser or cylinder into which the pipe end is free to work; unless this box is deep, and has a good taper at the bottom, and the part of the pipe fitting in the stuffing-box of brass and turned to fit, it is difficult to keep it quite tight. A better plan is to form the exhaust pipe with a good bend, so as to do away with the joint; or, if this cannot be done, fit a brass or copper *bellows* joint. In compound engines some trouble is often experienced from leakage through the drain cocks of the low-pressure cylinder, and since, when the engine is working under full speed, the pressure in this cylinder is below that of the atmosphere as a rule, to make these cocks at all effective, pipes should be led from them to the condenser.

Pump Rods.—The vertical single-acting pump in oscillating paddle-wheel engines is generally worked by a connecting-rod from a crank, or by means of a large eccentric, whose rod passes through a trunk cast with the bucket, and connected to a socket fitted to the bucket, and secured by a brass cap-nut underneath.

$$\text{Area of section of the rod} = 0{\cdot}01 \times \text{area of pump bucket,}$$

Or in case of a round rod

$$\text{The diameter of rod} = 0{\cdot}1 \times \text{diameter of pump.}$$

When two bolts are fitted to connect the brasses, &c., at the end,

$$\text{The diameter of each bolt in the body} = {\cdot}056 \times \text{diameter of pump.}$$

When the air-pump is of the single-acting type in screw engines, it is generally worked by means of a rod, either of one of the bronzes or brass rolled or iron cased with brass, having a tapered end fitting into the piston or bucket, and secured by a nut on the top side, or tapered the other way and secured by a cap-nut of brass (fig. 75) on the bottom side.

The diameter of the rod = $\cdot15 \times$ diameter of pump.

It is no unfrequent thing, though, in vertical engines, to find the pump worked by a connecting-rod and trunk as in paddle-wheel engines, the motion being derived from levers worked from the piston-rod crosshead in the same way as for pumps with fixed rods.

The arrangement of the pumps is various as regards detail, but for vertical screw engines is generally alike in principle, and may be divided into three systems—(1) Air-pump and circulating pump, both single-acting, and each worked separately by levers, &c., from a piston-rod crosshead; or air-pump and circulating pump side by side (the latter being either single or double-acting), and worked by levers from one piston-rod crosshead; or a single-acting plunger circulating pump inverted over the air-pump, and having a common rod, crosshead, levers, &c. (2) Air-pump and circulating pump worked each from a piston direct, or from the same piston-rod crosshead. (3) Both pumps worked by eccentrics on the crank-shaft, or by a crank-pin on the forward end of the crank-shaft. (4) The air-pump worked in one of the ways above described, and the circulating pump a centrifugal driven by an independent engine. (5) The air-pump detached and worked by an independent cylinder or engine, and the circulating pump centrifugal and worked by an independent engine. (6) The air-pump and circulating pump combined and worked by the same independent engine, the latter being centrifugal or reciprocating.

The first and fourth systems are the ones most generally adopted, the others being specialities of comparatively few engineers.

When the two pumps are worked independently of each other, each from a crosshead by levers, and are so arranged that in cases of emergency either can be used as an air-pump, the chances of breakdown are materially decreased; likewise the strain from the pumps is distributed, and their weight tends to balance the piston; but *per contra* the expense of working parts is increased, and the number of parts requiring attention and liable to accident is doubled; more space is required, and, in the case of compound engines, the low-pressure piston requires the greater counterbalance, so that it is an advantage to have both pumps worked from it.

If the two pumps are side by side, their rods are connected to a common crosshead, &c., thus leaving the space on the forward or after side of them free for the condenser, and admitting of more elasticity in its design; and, as is generally the case, the pumps are worked from the after cylinder, the condenser is easy of access, and there is plenty of space for the tubes to be drawn in a fore and aft direction.

The circulating pump inverted over the air-pump is a very convenient arrangement for small engines; but as the strains from the circulating pump have to be taken by the stand pipes from the pumps and the condenser top, it is found to give trouble in large engines; also, when the ship is light, and the valves out of order, it fails to pump well; but a large amount of space is saved by this plan, and it was at one time very frequently found in small ships.

Horizontal Pumps.—In the horizontal engine the air-pump was almost invariably worked direct from one of the pistons, although occasionally it was found more convenient to obtain the motion from one of the piston-rods of double-rod engines by means of a bracket or arm keyed to it; in either case the stroke of the pump is the same as that of the engine.

The air-pumps of horizontal engines are now often of the vertical type, single-acting.

The horizontal pump is nearly always double-acting, about the only case of a single-acting pump being that of Messrs. R. Napier & Sons in their old engines, when the pump-plunger formed a trunk in which the connecting-rod worked.

There were two forms of double-acting pump in general use, one being an ordinary piston of brass, packed with hemp gasket, working in a brass barrel, fitted in the condenser, and the other a hollow-plunger working through a brass stuffing-box, &c., fitted in the division-plate between the back and front sets of valves.

The advantages of the latter plan are, (1) more room is given for the valves (as the seats can be continued as far as the division-plate), without causing a large space between the head and foot valves. (2) The condenser-bottom or pump-box does not require to be bored out, as the stuffing-box can be easily fitted; while, in the former plan, the space for the pump-liner has to be bored out, and the liner turned to fit it, and afterwards secured in place. (3) The plunger will be about the same weight as the liner, thereby costing about the same from the foundry; but the work on it is principally plain turning, which is both cheaper and easier than boring, as in the case of the liner. (4) The packing can be renewed without disturbing the pump. On the other hand, the weight to be moved in the case of the plunger is greater than that of the piston-bucket (although it is sometimes supported by the water), and, in consequence, besides requiring more work to drive it, the wear is considerably more, so that in course of time it drops *out of line*, and is difficult to keep tight; the plunger also requires more room at the back and front of the condenser for its removal, which in small ships is often a serious consideration. Therefore, on the whole, the plunger system is the cheapest and easiest of design, while the piston-bucket works best, and is most convenient for examination.

The piston-buckets are made of bronze (in one thickness generally), stiffened by from 4 to 8 ribs; the circumference is either recessed down, so as to admit of gasket being coiled around it, or else it is formed like a stuffing-box, and fitted with a gland or junk-ring, secured by studs and nuts, which pass through lugs cast with it, so as to jam the packing tight after it has been placed in.

The Admiralty now, however, always specify the condenser and pumps to be wholly of brass, the former being of the same composition as the tubes, and the latter of the ordinary Admiralty bronze mixture.

Air-pumps are sometimes fitted with spring rings and junk-rings, similar to those of a steam-piston, but made of gun-metal instead of cast-iron; this, however, has been but rarely done, and then only in very large pumps, as in ordinary cases hemp packing serves the purpose very well, and is much cheaper. Vertical pumps will work well without packing; but, when packed, the packing is usually Manila rope or gasket in the mercantile marine, and bronze Ramsbottom rings in the Navy.

The diameter of the horizontal air-pump rod = $\cdot 15 \times$ diameter of the pump + $\frac{1}{4}$ inch.

The rods working in the pump are of gun-metal or Muntz metal; the Admiralty now prefer the former or rolled bronze, and have therefore struck the latter out of their specifications as being too soft for continued wear. When the rods are of large size, they are sometimes made of wrought iron, cased with gun-metal, but unless the rods are large and long, it does not pay to case them.

The rod fitting into the steam-piston is, of course, of wrought iron, and is connected to the brass rod by a box-coupling and cotters, the socket being formed with the iron rod, in order that the piston-bucket may be drawn out from behind for examination without removing the rod.

The pump-rod is usually fitted into the bucket in a similar manner to that of a piston-rod, the taper at the end being about 1 in 12.

Pump Buckets.—The following rules give the dimensions of an ordinary pump-bucket, of which fig. 75 is an example:—

$$x = 0{\cdot}3 \times \sqrt[4]{D} + 0{\cdot}15 \text{ inch.}$$

D is the diameter of the pump in inches.

The thickness of the disc when solid = $1{\cdot}0 \times x$.
" " " perforated = $1{\cdot}7 \times x$.
" " flanges at edge = $1{\cdot}1 \times x$.
" " metal around rod end = $1{\cdot}5 \times x$.
" " " the rim = $1{\cdot}0 \times x$.
" " packing = $1{\cdot}1 \times x$.
" " ribs = $0{\cdot}8 \times x$.
The breadth of the packing = $4{\cdot}0 \times x$.
" depth of bucket at the middle = $6{\cdot}0 \times x$.
" number of ribs, one for each 4 inches of diameter.

The bronze liner is usually from ½ inch to ⅞ inch thick, or its thickness $= 1{\cdot}1\,x - 0{\cdot}2$ inch.

This liner is let into the cast-iron bottom, and made to fit tight at both ends, facings on both liner and bottom being formed for that purpose; these facings are made with a very slight taper, so as to allow of a little *draw* on fitting in; the liner is then secured by brass screws passing through and through. The pump-barrels of vertical pumps are usually of bronze, ⅝ to 1 inch thick, secured to the foundation or pump bottom by flanges cast with them; the *head* valve-boxes, &c., are also borne on them, and secured by flanges in a similar manner.

Pump Valves.—The arrangement of the valves of an air-pump is of the highest importance, for, as has been stated, the efficiency of the pump to a very large degree depends on it; especially is this the case with modern compound engines, where but a small amount of water is pumped at each stroke. A triple-compound engine indicating 2000 horse-power uses about 7 cubic feet of water per minute, so that for a single-acting pump, with 100 revolutions of the engine, the amount per stroke is only 121 cubic inches; in practice, however, the pump fails frequently to take this amount, and so, until it accumulates behind the foot valves, no water enters the pump. Again, a double-acting pump is seldom so efficient at the front end as it is at the back, and, consequently, until there is a glut of water, the front ceases to pump. From causes of this kind trouble was often experienced with the double-acting pump of surface-condensers, and a remedy that has answered very well indeed in obviating this is to connect the space between the valves to the hot-well by a pipe about ·15 the diameter of the pump, having a suction or non-return valve fitted to it, so that the pump is to some extent continuously charged with water.

There are certain conditions that should be most carefully observed in arranging the valves, and they are—

(1) The foot valve should be in such a position that the water readily drains from the condenser into the pump.

(2) The valves should be so designed as to open with the least possible pressure under them, and still readily close on the return stroke of the pump.

(3) The flow from the condenser to the hot-well should be as direct as possible; that is, there should be as little change as possible in the direction, and no obstacles, so that the velocity generated at leaving the condenser should not be checked until the water is in the hot-well.

(4) The space between the head and foot valves of a double-acting and the head and bucket valves of a vertical single-acting pump should be a minimum, and all recesses, &c., where air can lodge or eddies can form, should be avoided.

To drain the condenser efficiently, the foot valve should be below its lowest level, and, when possible, the channel from it to the pump inclined so that the water naturally flows in that direction; all stiffening ribs, &c., should be placed on the outside, so as to avoid gutters in which the water can lie.

The foot valves can only open by the pressure under them being greater than that on them, and when, as in the case of surface-condensers, the head of water is not sufficient to do so, it depends entirely on the pump forming a better vacuum between the foot and head valves than there is in the condenser. Now, if there be a good vacuum in the condenser, it requires the full efficiency of the pump to produce a better one between the foot and head valves. At most there can be but little difference between the pressure in the condenser and that in the pump-chamber, and if the valves have much resistance in themselves they will cease to act. The full force of this is better appreciated when it is remembered that the pressure in the pump-chamber must vary from slightly above atmospheric pressure, when the head valves are opened, to below that in the condenser in order to open the foot valves. For this reason, if the space between the head and foot valves is more than one-fifteenth the capacity of the pump, it is impossible to obtain a good vacuum in the condenser. This clearance space is virtually reduced by filling it with water; without such virtual reduction very few horizontal air-pumps would be capable of maintaining over 20 to 22 inches of vacuum in the condenser.

The makers of the horizontal air-pump usually arrange the foot valves so that they are either on a vertical plane seating or on one inclined so as to allow of the valves opening easily; some engineers even invert the foot valves so that the water on them causes them to open so soon as the pressure in the pump-chamber equals that in the condenser. A brass box, having valves on five of its sides, and bolted to the condenser by the sixth, so as to cover the outlet orifice from the condenser to the pump, is sometimes used. In this way the clearance space is partly filled, and five valves can work in the space usually occupied by one; three of the valves are in vertical planes and open horizontally, the two others are in horizontal planes, one opening upward and one downward. Sometimes the inverted valve is adopted for foot valves; these valves were of india-rubber, circular and round-backed, so as to close quickly and keep tight on their faces.

The head valves should also be carefully arranged, for although not so difficult to place as are the foot valves, a little want of forethought may cause a bad working pump. They should always be so arranged that water lodges on them, so that in case they leak only water, *and not air*, escapes back into the pump; the valve seats and parts adjacent should be free from pockets underneath in which air can collect, and for this purpose they should be in the highest part of the pump-chamber, and if any part is partitioned off by a stiffening web or fillet, a communication should be made from its top to the underside of the head valves.

Since mineral oil has been used as the lubricant for cylinders, ordinary india-rubber ceased to be used for air-pump valves, owing to that oil being a solvent of the rubber. Many attempts have been made to manufacture a rubber which will withstand the oil, but none of them have been perfectly successful, and only a few have succeeded in preventing the rapid action noticeable on nearly pure rubber.

Vulcanite Valves.—India-rubber can, by a certain process, be converted into a hard black substance resembling ebony, and hence sometimes called *ebonite;* this substance is light and very strong, and quite impervious to oil, and has been used with great success for air-pump valves. The valves made of vulcanite (fig. 76) are generally circular and flat, strung on a stud through the middle, on which is a flat brass guard of the same diameter as the valve, and against which the valve lifts bodily. The wear of these valves is very little indeed.

Wooden Valves.—With the same object wooden valves have been tried, but not with the same success; the wood is not, as a rule, strong enough when light, and is destroyed by the combined action of the water and the blows on the seat.

Metallic Valves.—In the old days the air-pump valves were usually of brass and of large size, the foot and delivery valves being hinged or "flap" valves, and the bucket valve annular. The flap valve is still employed where the foot valve is not accessible or easily got at, and answers the purpose very well indeed; it should be of ample size, and have very little lift.

Coe & Kinghorn's Patent (fig. 77) consists of tongues made of very thin rolled sheet phosphor bronze. These tongues or flaps cover a grating in the same way as india-rubber, and are fitted with curved guards to admit of a gradual bend. These valves work very well, but great care is necessary in making and setting the guard, so that when the valve is open there is no change of flexure or angle on which the flap can work and gradually break.

India-rubber Valve (fig. 78) is a hog-back valve made of rubber of a quality to resist mineral oils, and a form which causes it to shut quickly and tightly on its seat; it is suitable for suction valves when placed horizontally.

Thompson's Patent (fig. 79).—A large number of small saucer-shaped valves, of phosphor or manganese bronze, cast very thin indeed, are sometimes fitted, and, being very strong and comparatively light, do the work very well. Single discs of thin sheet-brass or bronze have also been used for this purpose with success. Thin discs of brass are often used; they are inexpensive and work very well. Discs of fibre also answer very well.

The valve shown in fig. 80 is simply a thin, metallic disc without stiffening of any kind. When very little lift is given to this valve it works very well, and rarely gets out of shape sufficiently to be seriously inefficient. It is sometimes fortified by a second and a third thin disc, each one smaller than the one on which it is superposed, and having holes through it on a circle of less diameter than those in the disc below it.

In this way the lower disc permits water to flow around its edge and through the holes in it, and when closed the holes in it are covered by the next disc above it.

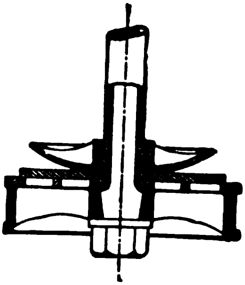

Fig. 75.

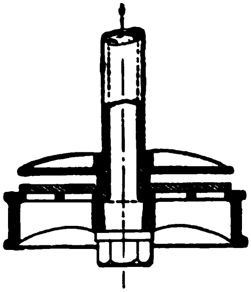

Fig. 76.

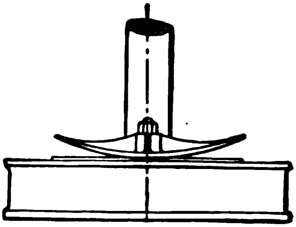

Fig. 77.

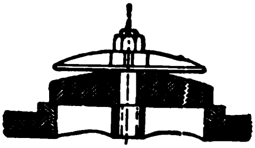

Fig. 78.

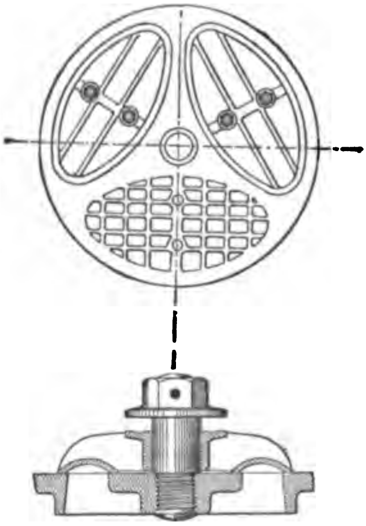

Fig. 79.—Annular Valve.

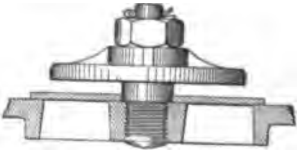

Fig. 80.—Flat Disc Valve.

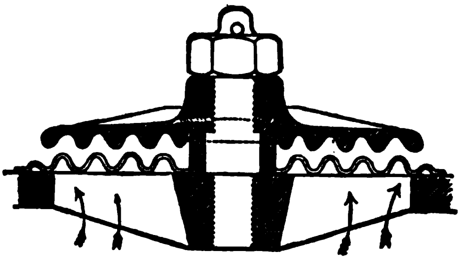

Fig. 81.

Figs. 75-81.—Air-pump Buckets, Valves, &c.

Beldam's Patent Valve (fig. 81).—In this case the valves are made of very thin sheet metal and stiffened by a series of circular corrugations. This valve has answered the purpose very well, for, while maintaining an even flat surface to cover the seat, it is elastic and can give under an unfair pressure and return to its normal form without permanent distortion.

Area through Valve Seats.—The area of opening past the valves depends on the size and velocity of the pump, and should not be less than will admit the full quantity of water for jet condensation at a velocity not exceeding 400 feet per minute. In actual practice the area was generally in excess of this. If the foot valves are large they will be sluggish in action; if they are small the velocity of water, &c., will be sufficiently high to raise the valves and keep them open by the energy of the particles striking them; this argument especially applies to the pumps of surface condensers. It may be noted that the vertical pumps of the old jet-condensing engines were frequently without foot valves.

If D be the diameter of the air-pump in inches, and S its speed in feet,

$$\text{Area through foot valves} = \frac{D^2 \times S}{1000} \text{ square inches.}$$

$$\text{Area through head valves} = \frac{D^2 \times S}{800} \text{ square inches.}$$

$$\text{Area through bucket valves} = \frac{D^2 \times S}{900} \text{ square inches.}$$

$$\text{Diameter of discharge pipe} = \frac{D}{35} \times \sqrt{S} \text{ inches.}$$

When the use of the "jet" is not possible, the area may be less than given by this calculation by 30 per cent., and the discharge pipe about half the diameter given by the above rule.

If the pump is without an air vessel or receiver, as is often the case with double-acting pumps, the diameter of discharge pipe should be 10 per cent. larger.

An air-pipe should always be fitted to the hot-well, as high up as possible, and its diameter should be ¼ that given above.

The bucket of a single-acting pump must be sufficiently large to admit of a valve area through it not less than given above. To obtain this, they are better designed with a short stroke. It is usual now and certainly advisable to deliver the water from the air-pump into a tank about 12 times the capacity of the pump; the delivery pipe is in diameter 0·7 that given above. The air-pipe of the tank should be about half the area in section. The tank should have an overflow pipe and a glass gauge. The suction for the feed pumps should be near the bottom and fitted with a strainer.

Circulating Pumps.—Two kinds of pumps are employed to circulate the cooling water in the condenser—the ordinary single- or double-acting reciprocating pump, and the rotary pump. The single-acting pump is usually fitted to small engines, and the double-acting to larger ones. The latter is preferable to the former—but more expensive. They are generally worked by the main engines, but sometimes by independent ones.

Single-acting Pump.—This is used in vertical engines, and is similar to the single-acting air-pump already described. It works very well, and, when provided with an efficient air-vessel, the flow is fairly steady. In very small engines a plunger-pump is used, and formerly inverted plunger-pumps were fitted to engines of considerable power. A good pet-valve, which will admit air to the pump and not allow the water to pass out, should be fitted; and likewise a pass-cock, which opens a communication between the delivery and suction, is a requisite; the former prevents noise by providing an air cushion for the water, and the latter checks the supply without straining the pump.

Double-acting Pumps.—These give a steadier flow of water, and cause less shock and strain on all the working parts, pipes, &c., than do the single-acting pumps; but even these should be fitted with pet-valves and pass-cocks. When a reciprocating pump is used for circulating water in a horizontal engine, it is always double-acting, for the same reason that the air-pump is so, and when this kind of pump is used for vertical engines of 150 N.H.P. and upwards, it should be double-acting.

Size of Circulating Pump.—The capacity of this pump depends on the quantity of cooling water and the number of strokes per minute.

Let Q be the quantity of cooling water in cubic feet, and n the number of strokes per minute, and S the length of stroke in feet.

$$\text{Capacity of circulating pump} = \frac{Q}{n} \text{ cubic feet.}$$

$$\text{Diameter} \quad ,, \quad ,, = 13{\cdot}55\sqrt{\frac{Q}{n \times S}} \text{ inches.}$$

Example.—To find the diameter of a double-acting circulating pump of an engine condensing 2 cubic feet of water per minute, and requiring 40 times the amount of cooling water; the stroke of the pump is 18 inches, and the number of revolutions 120 per minute.

Here

$$Q = 40 \times 2 \text{ or } 80 \text{ cubic feet; } n = 120 \times 2 \text{ or } 240.$$

$$\text{Diameter of pump} = 13{\cdot}55\sqrt{\frac{80}{240 \times 1{\cdot}5}} = 6{\cdot}4 \text{ inches.}$$

The size of the circulating pump is to a large extent dependent on the same conditions that determine the size of the air-pump, and may therefore bear a constant relation to the size of the air-pump; and since the size of the air-pump is often determined by the size of the cylinders, that of the circulating pump may be found in a similar manner. When the circulating pump is *single-acting*, the capacity should be 0·038 of that of the low-pressure cylinder, and when the circulating pump is *double-acting*, 0·021.

TABLE XXI.

Description of Pump.	Description of Engine.	Ratio.
Single-acting, -	Expansive 1½ to 2 times, - - - -	13 to 16
,, ,, -	,, 3 to 5 ,, - - - -	20 to 25
,, ,, -	Compound, triple, and quadruple, - -	25 to 30
Double- ,, -	Expansive 1½ to 2 times, - - - -	25 to 30
,, ,, -	,, 3 to 5 ,, - - - -	36 to 46
,, ,, -	Compound, triple, and quadruple, - -	46 to 56

When the air-pump is *double-acting*, the capacity of the *double-acting* circulating pump should be 0·52 of that of the air-pump, the double-acting circulating pump being more efficient than the double-acting air-pump.

Table xxi. gives the ratio of capacity of cylinder or cylinders to that of the circulating pump.

Circulating Pump Rods are made of the same materials, and in the same way as for the air-pump, and when possible the rods of both pumps are made identically alike, so that one spare rod serves for both.

Diameter of circulating pump-rod = 0·22 × diameter of pump.

When the pump is double-acting, or of comparatively long stroke, = 0·22 × diameter of pump + ¼ inch.

Circulating Pump Bucket.—When single-acting it is similar to that of the air-pump, and when double-acting it is simply a piston of brass.

Although it is usual to pack these pumps with hemp .gasket or bronze rings, there is no necessity for this, since the water flows freely into the pump by gravity, and the pump moves too quickly to allow of much leakage past the piston. Many engineers now dispense with packing, and simply make the piston a fairly good fit in the barrel; while others turn either a spiral groove, or a series of parallel grooves, on the edge of the piston, which has the effect of keeping the surface well lubricated and preventing leakage *when working*. The friction of the unpacked pump is considerably less than that of a packed one, and in fast-running engines this is no slight consideration. It may be taken now as certain that it is better not to pack the circulating pump of a fast-working screw engine, and packing is of doubtful advantage when the engine is slow-working, provided the pump is below the water-line.

Since brass and lignum vitæ work well together when lubricated with water, the piston of the circulating pump is sometimes fitted with a lignum vitæ ring, composed of several pieces let into a dove-tail groove. To allow for the expansion of the wood, the piston should be a slack fit when put into place new.

Circulating Pump Valves.—These are almost always of the best india-rubber, and of the quality known as "floating," from the fact of the specific gravity of rubber, with only such slight admixture of foreign matter as to render it usable, being less than that of water.

Valve Area.—The clear area through the valve seats and past the valves should be such that the mean velocity of flow does not exceed 450 feet per minute. It is not always easy to obtain so large an area, but when the velocity of flow is high the valves wear out more quickly, and the resistance of the pump is considerable, so that every effort should be made in this direction.

The pipes should be of such a size that the mean velocity of flow through them does not exceed 500 feet per minute when comparatively small, and long for their size; when large, and having fairly easy leads, the allowance may be as much as 600 feet. The suction pipe of a double-acting pump should be of the same size as the delivery pipe, but it may be considerably smaller when the pump is single-acting and below the water-level.

Diameter of Suction Pipes for circulating pumps may be obtained as follows, when the cooling water per I.H.P. per minute is not more than 8 pounds, and the velocity of flow as above:—

$$\text{Diameter in inches} = \sqrt{\frac{\text{I.H.P.}}{25}} + \tfrac{1}{4}\ \text{inch.}$$

This may be used for centrifugal pumps, as also for good double-acting pumps.

Let A be the area of the pump, D its diameter in inches, S the mean speed of movement in feet per minute; then

$$\text{Area past the valves is not less than } \frac{A \times S}{450}.$$

$$\text{Diameter of pipes} = \frac{D}{F}\sqrt{S}.$$

Suction	pipe of	small	double-acting	pump,	F = 22.
Delivery	„	„	„	„	F = 23.
Suction	„	large	„	„	F = 24.
Delivery	„	„	„	„	F = 25.
Suction	„	small	single-acting,	„	F = 26.
Delivery	„	„	„	„	F = 22.
Suction	„	large	„	„	F = 27.
Delivery	„	„	„	„	F = 24.

Both kinds of pumps should have air-vessels, and the single-acting pump should have one twice the capacity of the pump when possible, and never less than one-and-a-half times its capacity. When the water is *pumped through* the tubes, the doors of the condenser may be made with pockets, which serve as air-vessels in forming a cushion. Air-pipes should be fitted to the highest points of the waterways, when the water is pumped into the condenser, to allow the air to escape, so that it may run full, and always allow water to be in contact with the tubes.

Rotary Pumps.—There are one or two forms of rotary pump which have been tried for the purpose of circulating the cooling water, but only the centrifugal pump achieved perfect success.

The advantages of the rotary pump are—

(1) There are no valves, &c., to interfere with the flow of water, or to get out of order.

(2) Being easily worked by an independent engine of small size, it is usually so provided, and can be then started before the main engines are moved, and so keep the condenser cool during the process of "warming through" the cylinders, &c., and also while standing.

(3) Having this independent engine, the supply of cooling water is varied to suit the varying circumstances, and the power required to work the pump varies then with the quantity of water.

(4) The efficiency for low lifts is greater than that of a reciprocating pump.

(5) The supply of water is continuous, and enters the condenser without shock, thereby putting no strain on the castings, pipes, &c., beyond that due to the "head."

(6) It is easily placed in the engine-room, and the absence of a reciprocating pump worked by the engine, especially in a horizontal engine, admits of a better design and arrangement of condenser and air-pump.

On the other hand, the centrifugal pump is somewhat more expensive than an ordinary pump, and requires more attention when at work; the latter objection is, however, now considerably lessened, although only by increasing the former.

Centrifugal Pump.—This essentially consists of a wheel (fig. 82) having thin vanes as arms, which act on the water so as to give it a circular motion in a cylindrical case enclosing the wheel; this case is provided with an enlarged chamber around it, into which the water from the wheel is whirled, and from which it escapes through a branch tangential to it.

The principle on which this pump works is, that a particle moving in a circular path is under the action of two forces, one tangential, and the other normal to its path, or at right angles to the tangential one; when this latter force ceases to act, the particle moves away in a path tangential to the circle at the point where the retaining force ceased to act. In the centrifugal pump, the particles of water flowing into the centre of the pump are gradually put into motion and whirled round until, after a spiral course, they arrive in the outer channel, and there are retained in a circular course till they reach the outlet, where the retaining or normal force ceases, and they fly away tangentially through the outlet. The vanes of the wheel are sometimes enclosed between two discs of thin metal, to which they are attached, or they fit closely to the sides of the case in which they work.

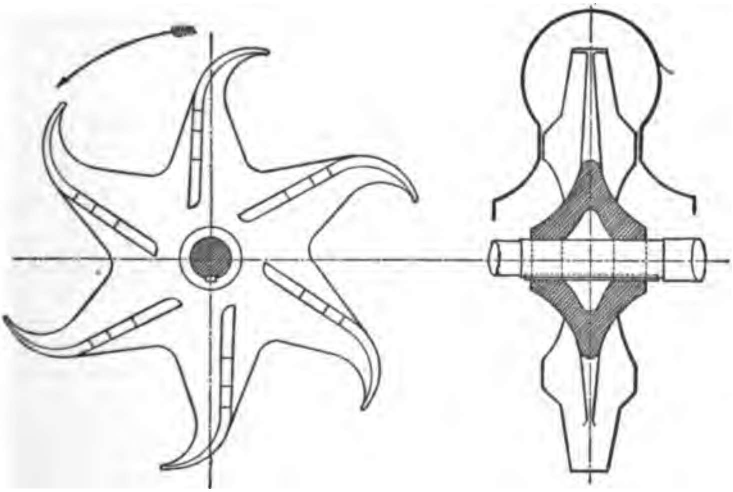

Fig. 82.—Wheel of a Centrifugal Pump.

Experience has shown that the best form of wheel is one whose vanes are sickle-shaped, as shown in fig. 82, and there is less resistance if the vanes fit the pump-case instead of having the enclosing discs, inasmuch as these latter have considerable friction on their surfaces, especially on that next the pump-case.

The outer passage, or *whirl chamber*, is usually formed like a snail, so that its sectional area gradually increases from nearly nothing to the full area of the discharge pipe.

The inlet pipe leads from the outer rim to the centre of the pump, having generally a passage on either side, so that the water is delivered on both sides of the wheel. For convenience these pumps are sometimes designed now with the inlet at one side only.

The diameter of the inlet and outlet pipes of a centrifugal pump for circulating purposes should be such that the velocity of flow does not exceed 400 feet per minute. Hence, if W is the quantity of water in gallons per minute,

$$\text{Diameter of pipes in inches} = \sqrt{\frac{W}{13 \cdot 7}}.$$

For triple and quadruple engines the following rule gives ample size:—

$$\text{Diameter of pipe in inches} = \sqrt{\frac{\text{I.H.P.}}{C}}.$$

$C = 20$ for all ships liable to be employed in the tropics.

$C = 24$ for ships employed only in temperate zones.

The diameter of the fan-wheel is usually from 2½ to 3 times that of the pipes, and always is such that the velocity at its periphery at full speed is in excess of 400 feet per minute, and generally from 450 to 500 feet. Hence, if R be the number of revolutions of the fan-wheel per minute, and *d* its diameter,

$$\text{The diameter in inches must not be less than} = \frac{1700}{R}.$$

$$\text{Breadth of blade at tip} = \frac{W}{30\,d}.$$

The blades are curved as shown, that the water on entry may be gradually set into circular motion, and caused to flow into the whirl chamber with as little force as possible, and to leave the fan also gently.

The size of the cylinder for driving the fan can be calculated from the usual conditions; in practice its diameter is generally $2 \cdot 8 \sqrt{\text{diameter of pipes}}$, and its stroke 0·25 the diameter of the fan.*

The fan should be always of bronze, and made as thin as possible in the blades; the spindle should be of bronze or steel cased with gun-metal. The Admiralty require the spindle to be cast solid with the fan. No doubt there is considerable danger of the fan working loose if not well fitted and secured to the spindle. This, however, can be done with care when a forged or other strong form of spindle can be used, whereas by the Admiralty method only a casting, with possible spongy places, is allowed.

As these pumps are sometimes required to pump out the bilges and ballast tanks, it is usual to fit some means of exhausting the wheel-case of air, so as to cause the water to flow up into it, as the fan has no sucking power, and will not draw until fully charged with water. Messrs. J. & H. Gwynne usually fit a small rotary exhauster, worked by means of a belt from the spindle; and other makers of these pumps fit a small steam-ejector on the same principle as Rodgers' vacuum blower. When no such special means is provided, the pump may be charged from the sea, and the sea inlet left slightly open until the pump begins to draw from the bilge or tanks. In the Navy the circulating pumps are arranged to draw from the bilges; for this purpose throttle-valves are fitted to the suctions, so that the sea supply may be instantly shut off and the bilge turned on, this method having been found the most successful for this purpose.

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* Large pumps should have compound engines with a low-pressure cylinder of the size thus calculated; the high-pressure cylinder should be half the diameter when the boiler pressure is 120 lbs. and upwards.

Feed-pumps.—The duty of the feed-pump is to supply the boiler with sufficient water to meet its wants. It is supplied from the hot-well, so that when there is a surface condenser its supply is fresh water, and the amount, under ordinary circumstances, is the same as that evaporated in the boiler; but owing to leakage and waste by blowing the steam whistle, or using an auxiliary engine which exhausts into the air, the quantity of water condensed is not always sufficient to make up for that evaporated. It is found necessary also occasionally to blow some of the water out of the boiler, to get rid of scum floating on the surface of the water, and this waste must be made good by a supply from the sea or elsewhere. The water from a jet condenser is very nearly as salt as sea-water, and this, on being evaporated in the boiler, leaves behind the salt, &c., so that unless some precautions were taken the boiler would in time become filled with solid matter. To prevent such a large deposit of salt, &c., in the boiler, it was customary to blow out some of the very dense water from the boiler at fixed intervals, and as the blow-off cock was situated near the bottom of the boiler a considerable amount of solid matter was thus got rid of; the quantity of water blown out was made up by an extra supply from the hot-well. Now, since the surface condenser might leak, or an accident might happen, whereby jet condensation had to be resorted to, the pumps of engines fitted with surface condensers were formerly made sufficiently large to do duty under such circumstances.

Sea-water contains about $\frac{1}{33}$ of its weight of solid matter* in solution; its density is measured by a hydrometer, which in the Navy is marked with degrees, so that when floating in pure water the zero point is at the surface, and when in clean sea-water it marks 10°. In the merchant service, engineers are accustomed to speak of the density by the number of ounces of solid matter to the gallon, sea-water containing 5 ounces to the gallon.

Gross Feed-water.—To find the gross amount of feed-water, it must be decided to what degree of saltness the water in the boiler may be safely worked. The amount of scale or salt deposited on the surface of the boiler *does not depend on the density of the water*, but *only on the quantity of sea-water evaporated in the boiler*; the boiler of a surface-condensing engine may be worked with perfect safety and with economy when the water is four times the density of sea-water, when the naval hydrometer registers 40° and the mercantile 20 ounces; indeed, the Boiler Committee appointed by the Admiralty in 1873 recommend such boilers to be worked at 45°, and those of jet condensing at 35°.

Net Feed-water.—The net feed-water is that quantity required to make up for what has been used as steam in the engine and its auxiliaries; let this be denoted by Q; let n be the number of times the saltness of the water in the boiler is to that of sea-water, then

$$\text{The gross feed-water} = \frac{n}{n-1} \times Q.$$

This is the amount of feed-water which must be pumped into the boiler when salt water only is used, to maintain the saltness of n times that of the sea.

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* On the British coasts the sea-water contains only about $\frac{1}{40}$ of its weight of solid matter in solution.

If the pumps were only of sufficient size to pump this amount of water, a considerable time would elapse before the boiler would be filled to the working level after "blowing off"; to meet this objection it is usual to make *each* feed-pump capable of pumping twice this quantity, therefore

$$\text{Quantity of water for each pump to supply} = \frac{2n}{n-1} \times Q.$$

Example.—An engine condenses 2 cubic feet of water per minute in a jet condenser, the stroke of the feed-pump is 18 inches, and the number of strokes 120 per minute. The density of the water in the boiler is not to exceed $\frac{3}{32}$ (30°, or 15 ounces to the gallon). Here

$$\text{The gross feed-water} = \frac{3}{3-1} \times 2 = 3 \text{ cubic feet.}$$

$$\text{The quantity of water to be pumped} = \frac{2 \times 3}{3-1} \times 2 = 6 \text{ cubic feet.}$$

$$\text{The capacity of the pump} = \frac{6}{120}, \text{ or } 0{\cdot}05 \text{ cubic foot.}$$

And since the stroke is 1·5 feet—

$$\text{Area of plunger} = 0{\cdot}05 \div 1{\cdot}5 = 0{\cdot}0333 \text{ square foot.}$$
$$= 0{\cdot}0333 \times 144, \text{ or } 4{\cdot}8 \text{ square inches.}$$

Therefore, diameter of plunger is 2·5 inches.

Since a surface condenser supplies pure water for feeding the boiler, and the waste is now made up with fresh water, either carried in the tanks or distilled from sea-water, there is not the same need for blowing off the boilers, the feed-pumps may be very much smaller than when jet condensation is practised, and are generally of such a size when worked by the main engine that *each* is capable of delivering three times the *net feed-water*, assuming them to have an efficiency of 1·0; the pumps, when both are working, can then deliver six times the net feed, which is sufficient to satisfy the demands.

If Q be the quantity of net feed-water in cubic feet, l the length of stroke of feed-pump in feet, and n the number of strokes per minute,

$$\text{Diameter of each feed-pump plunger in inches} = \sqrt{\frac{550 \times Q}{n \times l}}.$$

If W be the net feed-water in pounds—

$$\text{Diameter of each feed-pump plunger in inches} = \sqrt{\frac{8{\cdot}9 \times W}{n \times l}}.$$

The following empirical formula will give such sizes as will be found in practice, and which will closely approximate to those given by the above rule:—

$$\text{Capacity of each feed-pump} = \frac{\text{capacity of cylinder}}{C}.$$

The following are the values of C when the engine is surface-condensing:—

Terminal pressure under 25 lbs., ... C = 220.
„ „ 20 lbs., ... C = 250.
„ „ 15 lbs., ... C = 320.
„ „ 12½ lbs., ... C = 380.
„ „ 10 lbs., ... C = 440.
Compound engines generally (taking L.P. cylinder only), C = 400.

The net feed-water in cubic feet per stroke is approximately = area of piston in inches × stroke in feet × absolute pressure at release ÷ 3,125,000; or,

Net feed-water in pounds per stroke approximately = area of piston in inches × stroke in feet × absolute pressure at release ÷ 50,000.

Example.—To find the net feed-water in cubic feet for an engine whose cylinder is 50 inches diameter, and length of stroke 2 feet, the pressure at release being 12 lbs., and the revolutions 100 per minute.

$$\text{Net feed-water } \textit{per minute} = 200 \times \frac{1963 \times 2 \times 12}{3{,}125{,}000} = 3{\cdot}01.$$

All engines over 80 N.H.P. should have two feed-pumps, each capable of supplying the boilers when the engines are at full speed; and each pump should be so arranged that it may be worked quite independently of the other, and easily put out of action when not required. This latter condition is seldom complied with in practice; but if it were so there would be then a spare pump, and only one in danger of derangement when the engine is at work.

In the horizontal engine the pumps are worked either from the crosshead, or direct from the piston, by means of a rod, consequently they have the same stroke as the piston; the former plan is preferable as being cheaper, and avoiding an additional stuffing-box in the cylinder. The feed-pumps of vertical engines are usually moved by the same parts which move the air and circulating pumps. They are sometimes fixed to the crosshead of these latter pumps, and sometimes driven from studs in the sides of the rocking levers. When there is only one set of rocking levers this latter plan should be adopted; for by placing one pump on each side of the lever centre they deliver alternately, and give a steady flow of water. In the Navy the use of feed-pumps worked by the main engines is entirely discontinued, and in the mercantile marine it is fast becoming the practice, especially in large ships, to feed the boilers by an independent pump, and generally a self-regulating one. In fact, with all fast-running engines the feed-pumps are better detached from them and worked with absolutely independent automatically-controlled engines, so that the supply of water is steady and not intermittent, and exactly in accord with the demand. The usual plans are the same in principle, though differing in detail. The water is delivered from the hot-well to a receiving tank by the air-pump in the mercantile marine, by a special pump in the Navy. The feed-pumps, two in number and in duplicate, draw from this tank at a uniform rate under normal circumstances, but if the supply in the tank fails the pumps are slowed down and stopped by a float arrangement connected to their stop valves. If the feed is wholly or partially shut off in the boiler-room the extra load on the pumps slows them down, and eventually stops them if "feed" is shut off. As a rule, only one of these pumps works at a time, and is capable of supplying two to three times the *net feed* when working full speed.

Relief Valves should be fitted to each pump when worked by the main engine, when they are so arranged that each may work separately; when they both deliver into a common pipe, one relief valve is sufficient; it should be loaded to 1¼ times the boiler pressure.

Valves and Valve-boxes should always be of bronze; and since the seats as well as the valves wear out rapidly, they should be made separately from the box casting. The valves should have a seating area equal at least to 20 per cent. of the valve, and it is better to make them flat rather than conical. Some engineers use brass balls fitting into conical seats; since they are constantly changing their position on the seats these balls wear very slightly and keep very tight. When the boiler pressure did not exceed 30 lbs., the valves of the feed-pumps were usually made of india-rubber; and American engineers still employ this material notwithstanding the increased pressure. If the rubber is thick and capable of withstanding the action of mineral oil, no doubt it will work well; but so much reliance cannot be placed on it as on the metal.

Bronze valves (fig. 79) make a noise when working, owing to the quick return to their seats on the pump ceasing to deliver; attempts have been made to reduce the noise and wear in various ways. The noise of the valves may be stopped by loading them with light springs made of steel and plated, or of hard brass. Several small valves with little lift is found to be more satisfactory than one large one; for example, to obtain sufficient area through the seats of six valves to take the place of one 4 inches in diameter, each would be 1⅝ inches, giving the sum of the perimeters 30·6 inches against the 12·56 inches of the 4-inch valve. It follows, then, that if the single valve lifted half an inch, the small one need only lift one-fifth of an inch. Such small valves must be made of forged Stone's or manganese bronze, carefully formed so that there is no groove or nick to form a starting point for a crack, as the tails quickly break off by the "jar" on seats and stops.

Air-vessels.—Each pump should be furnished with an air-vessel which may serve the double purpose of providing a cushion, and collecting the free air from the feed-water, which, when sea-water is used, is the active agent in producing corrosion when admitted to the boiler. To increase its usefulness in the latter capacity, a fine grating should be fitted to the inlet orifice, which will "spray" the water, and so separate the air, and a relief valve should be fitted to the top of the vessel, loaded to a pressure slightly below that obtaining when the pump is delivering at its maximum rate, but arranged to close when the vessel is nearly full of *water*. Air-vessels, however, are practically useless with the high pressures of steam now employed, except as a means of getting rid of gases and air.

Feed-pipes.—The pipes leading to and from the feed-pumps should be such that the velocity of flow does not exceed 500 feet per minute, and small pumps should have larger pipes in proportion, so that the flow through them does not exceed 400 feet.

Since the amount of water actually flowing through the pipes is generally very considerably less than the pumps are capable of discharging, the velocity is seldom more than half the above allowances; but as the pumps do occasionally deliver their full amount, the pipes must be large enough for that purpose.

If d is the diameter of the feed-pump plunger, and s its mean velocity in feet per minute, then

$$\text{Diameter of feed-pipe} = \frac{d}{20}\sqrt{s} \text{ for small pumps}$$

and

$$\text{Diameter of feed-pipe} = \frac{d}{23}\sqrt{s} \text{ for large pumps}$$

and

$$\text{Diameter of feed-pipe} = \frac{\sqrt{\text{I.H.P.}}}{15} + 0{\cdot}25 \text{ inch.}$$

Example.—To find the diameter of the feed-pipes for a pump whose diameter is 6 inches, and the length of stroke 2 feet, worked from the levers of an engine, making 60 revolutions per minute. Here $s = 2 \times 60 \times 2$, or 240 feet.

$$\text{Diameter of pipe} = \frac{6}{23}\sqrt{240}, \text{ or 4 inches.}$$

Example.—To find diameter of feed-pipe for an engine of 3600 I.H.P.

$$\text{Diameter of pipe} = \frac{\sqrt{3600}}{15} + 0{\cdot}25 \text{ inch} = 4\tfrac{1}{4} \text{ inches.}$$

If there are two pumps which deliver alternately, the pipes will be the same size throughout; but if the two pumps *may* deliver at the same time, the pipe beyond the junction of the two from the pumps must be nearly double the sectional area of one. As the resistance of pipes is due greatly to friction at the surface, and will consequently vary as the diameter, while the area of section varies as the square of the diameter, the resistance in the single pipe will be considerably less than the combined resistance in the two, and for this reason the sectional area may be less. In practice this area may be 0·8 of the combined area of the two. Hence, when there are two pumps delivering together—

$$\text{Diameter of main pipe} = 1{\cdot}265 \times \text{diameter of branches.}$$

If there were two pumps, as in the last example, delivering together, the diameter of the main pipe would be 4 × 1·265, or 5·06 inches.

"Pet Valves."—Although it is prejudicial to admit air to the feed-water, it is sometimes necessary for the good working of the pumps to allow a little air to enter between the valves to form a cushion. For this purpose "pet" valves are usually fitted.

Feed Tank.—To avoid any waste of water through the overflow or air-pipe of the hot-well when the feed-pumps are temporarily stopped, it is the rule to provide a tank, into which the water is discharged from the hot-well, and from which the feed-pumps draw. Such an arrangement is very beneficial in all engines, and especially in those having small hot-wells.

Feed-pump Rod.—This is of iron, and, in the case of hollow plunger pumps worked from a pin having a circular motion, it is jointed within the pump. As the plungers of vertical pumps are seldom without water in them, and difficult to empty, the joint should be such as will work with water as a lubricant. This is accomplished by bushing the joint with lignum vitæ, and casing the pins with brass; or white-metal (Fenton's) bushes, with iron pins, will do. The rod-end within the pump should be galvanised, as otherwise it often becomes corroded.

When the rod is long and of iron, p being the boiler pressure,

$$\text{Diameter of feed-pump rod} = \frac{\text{diameter of plunger}}{40} \times \sqrt{p} + 0{\cdot}6.$$

If of brass, divide by 35 instead of 40.

When the rods are short, the diameter may be 0·7 of that given by the above rules.

The plungers, valve-boxes, and valves are always of best bronze, and the Admiralty require the pump-barrel or case to be of bronze, but in the merchant service this is generally of cast iron. The same remark applies to the air-vessels, escape-valves, &c. The capacity of the air-vessel should be from 1·5 to 2 × the capacity of the pump.

Bilge Pumps.—These pumps, which are for the purpose of freeing the bilge of water, are somewhat similar in construction and method of working to the feed-pumps, and in engines with jet-condensers are of the same size; since smaller feed-pumps are required with surface condensation, this old rule does not hold good. Although the Board of Trade and Lloyds require that all steam ships shall have two bilge pumps worked by the main engines, except in the case of engines under 70 N.H.P., they do not specify the capacity of these pumps. There is no basis of calculation for the size of these pumps, and it is generally at the caprice of individual engineers, many of whom still adhere to the old practice of making them of the same size as the feed-pump. They may be, however, made to the following rule with advantage:—

$$\text{Capacity of bilge pump} = \frac{\text{capacity of cylinder}}{350}.$$

The plungers are usually of bronze, but in the merchant service are often of cast iron, and as this is harder, especially when the hard skin is formed by rubbing, they wear longer when of this material. When of cast iron, the neck and gland bushes should be of Fenton's metal, to prevent corrosion when not at work.

In the Navy the bilge pumps are, like the feed-pumps, quite detached from the main engines; in the mercantile marine, when the engines are run at a high number of revolutions, the same practice is followed; but when the engines are not quick running the bilge pumps are, as formerly, worked from the main engines. The Admiralty have a pump set apart for the bilges in each engine-room, and this is generally done in passenger steamers. There is also in each engine-room another pump which may be used for pumping out the bilges or for discharging water on deck; or one to do this service and the other to pump from the sea on deck or from the sea, and tanks to the boilers.

In the mercantile marine, the valve-boxes are of cast iron, and the valves usually hinged "clacks," capable of easy removal for cleaning. The covers of these boxes should be so made that they may be easily and quickly removed and replaced, and for that purpose are sometimes hinged, and, better still, held down by two hinged bolts fitting into recesses in the cover.

The Board of Trade requires that one bilge pump shall be arranged to draw water from the sea, and pump it on deck in case of fire. When this is done, the suction pipes should be fitted with a three-way cock, whose plug has only one port, so that it cannot be open to the sea and bilge at the same time, and so flood the ship. When the pipes are very large, it is not always convenient to fit a cock; but a double valve-box with self-acting non-return valves is substituted; when possible, the cock is the better and safer plan.

Directing Boxes.—It is required that the bilge pumps shall draw from each compartment of the ship. For this purpose, the suction from the pump is connected to the top of a box containing a series of valves by opening any one of which a communication is made to a separate compartment; the cover of each valve should have on it a label signifying to which compartment the valve opens a communication. These directing boxes should be placed in such a position, that they are easily got at, and above the floor plating when possible.

Mud Boxes.—Between the directing box and the pump should be fitted a box with a strainer, which shall intercept such solid matter as would derange the pump valves, if allowed to enter among them. They should have covers similar to those of the pump valve-boxes, and placed in such a position as to be easily got at.

Sanitary Pump.—In passenger steamers, it is usual to have a pump of about a half or one-third the capacity of the bilge pump, which can be put into gear, and worked by the engines, to discharge water on deck for sanitary purposes.

Hand Pump.—A pump arranged to be worked by manual labour, was usually fitted in all but very small ships, for the purpose of filling the boilers, clearing out the bilges, and pumping water to clean the decks when steam is not up. This pump was required by the Board of Trade Regulations made before the days of donkey pumps, and when the boiler pressure was only a few pounds over the atmosphere. The Board of Trade does not now insist on this useless apparatus. As a means of feeding the boilers by hand when under steam it was long ago quite useless.

The Admiralty used to require that the hand-pump may be worked by the engines. This can be effected by having a stud in a sliding block, on the end of the shaft, or on the levers of vertical engines; the sliding block is operated by a screw, so that it may be moved from the centre where there is no motion, to a point which will give the stroke required. The connecting-rod may have a gab to take the pin, or be permanently attached to the pin and made telescopic, with a cotter or set screw to secure it when required to work the pump.

The valves of the sanitary and hand pumps should be similar to those of the bilge-pumps.

Ejectors.—In the Navy, as in the merchant service, steam ejectors are often used for clearing the engine and boiler-room bilges. They are in principle very like the well-known injectors, and without valves or other internal fittings, liable to derangement from ashes, dirt, &c. They are started and stopped by merely opening and shutting a cock or valve, and require no attention when working, and are, consequently, most suitable for such a service.

CHAPTER XIII.

VALVES AND VALVE-GEAR.

STEAM was admitted to and released from the cylinders of the early land engines by means of conical valves, operated by tappet gear in such a way that the steam valve was suddenly opened and as suddenly closed at the proper times, and the valve which allowed the steam to escape to the condensers and closed before steam was admitted worked with the same precision and by the same methods. Such an arrangement permits of a high state of efficiency for the steam, but is open to the objection that motion so sudden is liable to cause much wear and tear of the working parts. In those early days the pressure of steam was only a little above that of the atmosphere, and the number of strokes per minute comparatively few, so that leakage past the valves and the wear of the tappet gear was not so great as might be expected; and, on the whole, the early engineers had every reason to be satisfied with their valve-gear.

That such gear can be made to work well and give general satisfaction is evident from the fact that it is still employed in large pumping engines, which work at 60 lbs. pressure and move at much higher speeds than formerly obtained, and in large electric generating engines.

Modifications of this form of valve and valve-gearing have been adopted for marine purposes, but not with that degree of success, at least in this country, as to commend themselves to engineers for extended adoption. In America, however, they are very generally employed for marine engines, and most successfully so in the large beam engines of river and lake steamers.

Such gearing was impossible in the locomotive engine on many grounds, so that an entirely different valve was adopted to suit it. This valve differed from previous valves in many ways, but chiefly in respect to its motion, which was a sliding one. From it this form of valve is called a *slide*-valve, and frequently, when in its simple form, *the locomotive slide-valve*, to distinguish it from the long and short D valves and piston-valves so generally used in the engines on land and sea in the early part of the nineteenth century, which also had a sliding motion.

The locomotive slide-valve (fig. 84), in its simple and extended forms, is the one most generally used in the marine engine. It consists essentially of a rectangular block having a central cavity, the flat bars between the outer edge at each end and the central cavity being sufficiently broad to cover the cylinder ports, and when in its mean position both ports are covered. The amount by which the outer edges overlap the ports when in the mean position is called the *lap*, and the amount by which the valve is open when the piston is at the commencement of its stroke is called the *lead;* the space through which the valve is moved during half a revolution of the engine is called the *travel*. The amount by which the inside edges of the valve overlap the port is called the *inside lap*, and when, as often happens, instead of overlapping inside, the port is slightly open to the cavity of the valve, the valve is said to have *negative inside lap*.

The early locomotives made by Stephenson had little or no lap, and little or no lead. Timothy Hackworth, by giving the valves of his engines lead and lap, effected an earlier cut-off, and consequently obtained expansion of the steam, thereby saving the locomotive from threatened failure, and making it a commercial success. The effects of *lead* and *lap* will, however, be shown later on.

The Long and Short D Valves spoken of are slide-valves whose cross-section is shaped like that letter, the flat part bearing on the cylinder face, so as to cover the port, and the curved back fitting against elastic or metallic packing to prevent the passage of steam past it. Fig. 83 shows this form of valve, and it will be seen that it differs from the locomotive slide-valve in not covering the exhaust-port at the sides, and by providing for passing of steam from one end to the other *through* itself, instead of being surrounded by steam. This kind of valve is practically relieved from the pressure on the face due to the steam, and therefore should have less resistance to motion than the slide-valve; but in practice the pressure from the packing was excessive, and even then it was very difficult to keep steam-tight, especially in the corners at the sides; it was consequently discarded in favour of the piston-valve or locomotive valve.

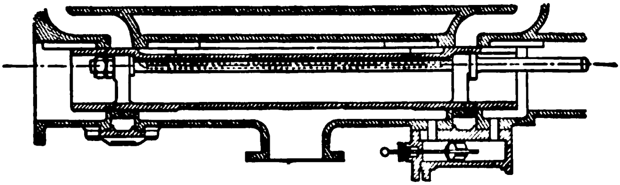

Fig. 83.—Long D Slide-valve.

Seaward's Valves.—To get a more perfect arrangement of cut-off and release than is possible with one valve, Seaward fitted a separate valve and ports for steam and exhaust, and to avoid the large amount of *clearance* which such an arrangement would entail, he used four valves, a steam and an exhaust-valve for each end of the cylinder, and by placing the ports close to the ends of the cylinders, reduced the passages to a minimum. Each valve was simply a flat plate, and worked by cams on the shaft, and however early the steam-valves cut off, the exhaust-valves always opened just at the end of the stroke. This arrangement, although admirable in many ways, has succumbed on account of complication of gear and multiplicity of parts; and, notwithstanding attempts to modernise it to suit the compound engine, it is now a thing of the past.

Common or Locomotive Slide-valve.—Fig. 84 shows the modern form of the valve, and it is so well known as to require no further description. So long as the cut-off is later than half the stroke of the piston, this valve answers very well for engines of moderate size; but when an earlier cut-off is required, sufficient opening to steam can only be obtained by excessive travel, early leads, or very broad ports, neither of which is desirable when it can be avoided. Large travel of valve means large power to drive it; broad ports produce the same result by increasing the area, and consequently the load on the valve; and early *leads* are apt to produce severe shocks on the rods, framing, &c., and to unduly check the piston.

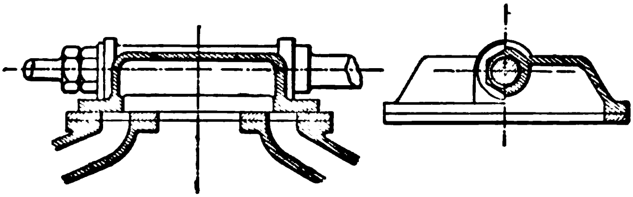

Fig. 84.—Common Locomotive Slide-valve.

Trick Valve.—Fig. 85 shows an ingenious plan for obtaining a double opening to steam by means of a passage around the valve, the entrance to which is at the end, remote from that at which steam usually enters, and whose exit is through the lap or cover of the valve. This invention is due to Herr Trick, and has been used on the Continent in locomotive and other fast-moving engines. It will be seen that the effective surface of the valve exposed to steam pressure is considerably reduced below that of the common slide-valve.

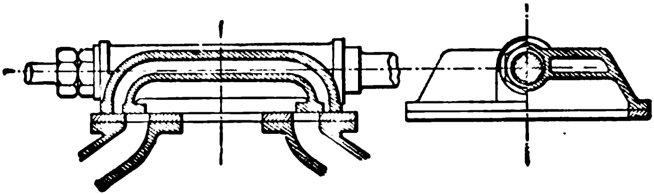

Fig. 85.—The Trick Valve.

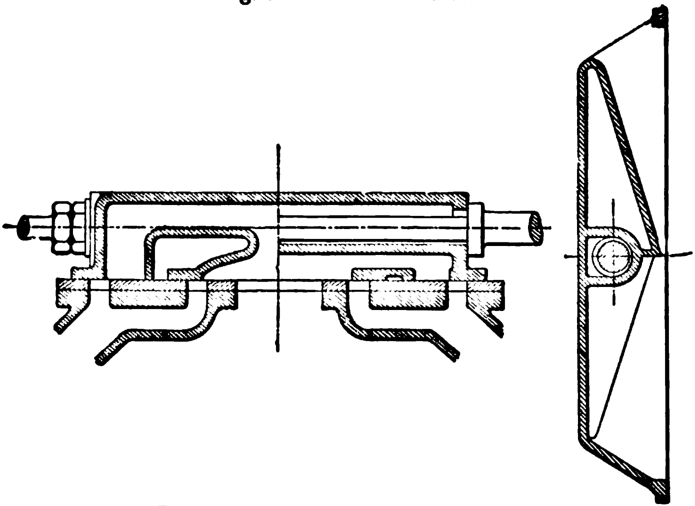

Fig. 86.—Common Double-ported Valve.

Double-ported Valve.—This valve (fig. 86) has a system of ports and passages which is added to the locomotive slide-valve, to allow of admission and emission of steam through a second port in the cylinder face, so that with the same travel as the common valve, there is double the area of opening for steam, and double the area for exhaust. This form of valve is very generally adopted for both the medium-pressure and low-pressure cylinders of compound engines of large size, and for the low-pressure cylinder of even very small engines. Care must be taken in designing such a valve, that there is the requisite area for steam to enter into the cavity leading to the steam port, and also that there is ample room at its back for the exhaust steam to pass from the outer port of the cylinder.

Treble-ported Valves.—To obtain still larger opening for steam, the lap of the double-ported valve is extended so as to cover a third port at each end; a portway is made through the cover lap, which admits steam to the *second* port of the cylinder, while steam enters the third port of the cylinder through the opening beyond the valve in the usual way. But since the length of face and valve required for this form of valve is very nearly as much as if it had a port and passage to admit of exhaust, it has been discarded in favour of a treble-ported valve, similar in all respects to the double-ported. Some very large engines have four-ported valves, similar in design to the double-ported. Such valves are very large and heavy, and require a large amount of power to move them; but they work very satisfactorily, and so far have not any successful rivals for steam of low-pressure. When used for the medium-pressure cylinder of compound engines, both treble- and double-ported valves require relief plates or frames to diminish the large area they expose to steam pressure.

The Travel of Flat Valves should be such that their mean speed is not more than 200 feet per minute for the H.P. cylinder, nor 250 for L.P. cylinders.

Hackworth's Valve.—Fig. 87 is an illustration of the valve invented by the ingenious Mr. J. W. Hackworth, and which gives three openings to steam for one cylinder port, and that with only a comparatively small area, unbalanced by steam pressure. The valve is a very simple casting, and although somewhat long is not so much so as the usual treble-ported valves. It is designed to be worked by a very ingenious arrangement of gearing, having only one eccentric, and capable of cutting-off at a very early period of the stroke, without undue cushioning, or too early release, as well as reversing.

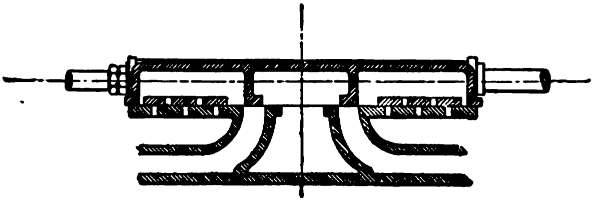

Fig. 87.—Hackworth's Patent Valve.

Piston Valves.—No system of relief frames or plates has yet been tried which has given universal and entire satisfaction; some indeed produce more resistance than they have been designed to reduce, and the best cannot be depended on for any very long period when exposed to the temperature of high-pressure steam. The area of opening of port is restricted when only a common locomotive slide-valve is used, and its extensions magnify the evil which relief frames are supposed to cure. It has been stated that circular valves of the *mushroom* type do not work well in fast-running engines, although they give a good opening to steam. To combine the advantages of the two systems the piston valve is designed (fig. 88). The port area is nearly three times that of a flat valve of the same dimension transversely, and the pressure on the sides due to the steam is nil. Essentially, the piston valve consists of two pistons, the face of each being equal in length to the bars of a locomotive slide-valve, and connected by a rod. These pistons are fitted into a cylindrical chamber having ports corresponding to those in the cylinder face; the faces of the piston cover these ports, and have the same amount of lap, &c., as a common valve. Steam is sometimes admitted outside the pistons, and it exhausts from the cylinder into the space between them, and from there into the exhaust passage in the usual way. But the piston valve permits of the steam being between the pistons and the exhaust steam passing their outside. In this case the valves are said to be with "inside cut-off." Piston valves were used very early in the 19th century, especially in marine engines, and found to be superior to the "Long D" and "Short D" valves; they were always of the "inside cut-off" type, and then, as now, had the advantage of confining the steam of high pressure to the circular casing surrounding the middle of the valve, so that no joint or stuffing-box was exposed to its action. This quality is to-day, with steam of really high pressure and high temperature, of great value, especially when the steam is rendered very dry by superheating, &c.

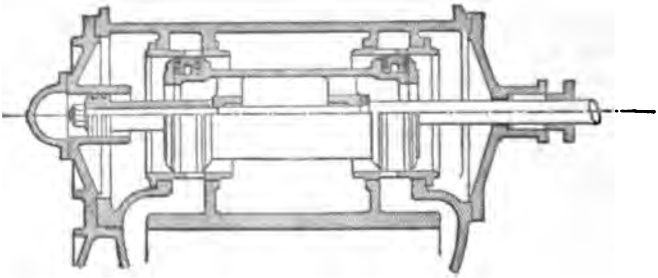

Fig. 88.—The Piston Slide-valve.

When the pistons are sufficiently large they are connected by a pipe or hollow casting (as shown in fig. 88), through which steam can pass from one end to the other; if this cannot be accomplished, the two ends of the valve-case are connected by a pipe cast with or connected to it.

Small engines, when fitted with such valves, have them in their simple form, the pistons being plain bronze or cast-iron discs of the required thickness, generally cast in one piece. Such a form would suit all sizes of engines, if always working at full speed; but when standing or running slow, the leakage past the valve, when it was worn, would soon be so considerable as to cause serious loss, besides making the engine very unhandy. To avoid this it is usual to pack the pistons much in the way that ordinary pistons are packed, except that the junk-rings and flanges are chamfered away, and the packing rings are made to project from them so as to allow free passage to the steam. The spring rings are made of strong cast iron, and since, owing to the very slight velocity at which the valve moves, the wear is small, the rings should have very little *set* or spring. When steam of high pressure is in contact with the valves of large size the rings should be restrained from pressing on the walls, as in the case of pistons (p. 158). The liners in the valve-box are usually made of cast iron, fitted tightly in and secured by flanges.

There are *diagonal* bars across the ports to act as retaining guides to the packing rings; these bars are usually from ¾ to 1¼-in. broad, and take away about a third of the gross portway. The passage way around the liner must be so designed as to allow due area of section for the passage of steam; and to economise space, and reduce the clearance space, they are eccentric to the liner and valve. To avoid the chief defect in these valves—viz., large clearance space, the valve should be long so that its ports are nearly in line with those at the cylinder bore.

Piston valves are now very general, and experience of them has given the necessary confidence, even for their more extended use. For steam of a pressure over 100 lbs. they have become a necessity, and they may be used with advantage for the high-pressure cylinder for pressures down to 75 lbs. Some manufacturers use them for the low-pressure cylinder, but few engineers will care to incur the expense of them for a purpose where they are generally quite unnecessary. They are, however, necessary for the fast-running engines, such as fitted in torpedo boats, destroyers, cross-channel passenger steamers, &c., where safe continuous running is of more importance than extreme economy. They also have advantages for very large engines, when the port area is so great that two valves are required to avoid cutting away the cylinder ends from the body.

Relief Frames.—The resistance of a slide-valve is almost wholly due to the friction on its face, and the greater the pressure on the valve the greater will be the friction. This pressure is due to the fact that that on the back of the valve is only very partially balanced by that on the front. The common locomotive slide-valve may, during certain portions of its stroke, have no pressure at all acting so as to tend to press it off its face. Just at cut-off it is relieved to the extent of the area of one port, but this relief decreases as soon as the steam in the cylinder expands: and if it exhausts before opening at the other end for steam, its whole area is exposed to the full amount pressing it to its face. It will be seen from this that the resistance is ever varying, and this statement is true also of double- and treble-ported valves, so that if a *definite and fixed* area of the back of the valve is so covered as not to be exposed to steam pressure, it does not meet the case; this, however, is the usual way in which the valve is relieved, the area being such that the valve is always pressed to its face. An exception to the rule is seen in the relief arrangement of the valves designed by Messrs. J. I. Thornycroft & Co. These valves have a face on their back similar to their front, with recesses corresponding to the ports; on the back of the valve there is a plate, which has ports like those on the cylinder face, bearing gently, so that, in one respect, it is as evenly balanced as a piston valve.

Double Valves.—Fig. 89 shows an ingenious arrangement by which one valve is made to balance another by attaching one to the other, and as the two valves operate for the same purpose, each may be half the size of an ordinary valve.

The two valves are faced up, and so set that they are a slack fit between the two faces when cold. The port-ways are, as shown on the diagram, bolted to the main casting on each side, and from their form the pressure

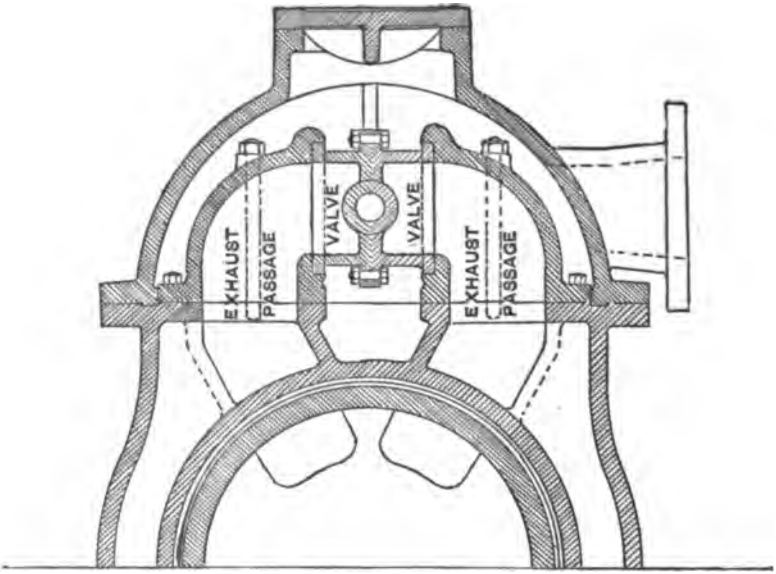

Fig. 89.—Self-balanced Valve.

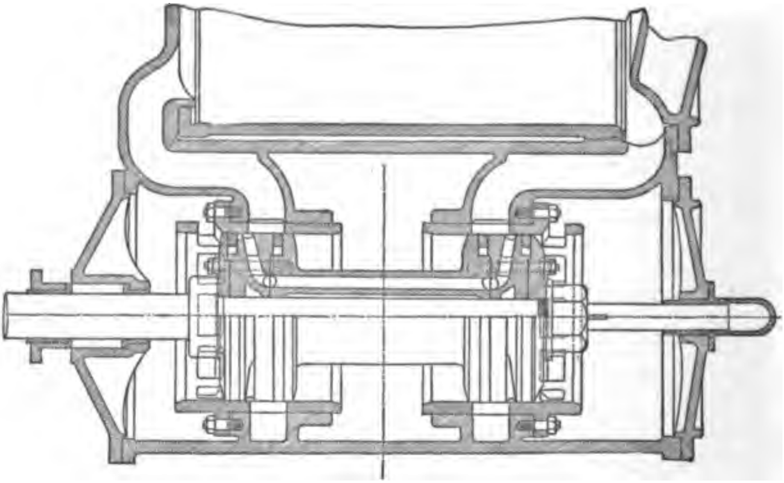

Fig. 90.—Thom's Patent Piston Valve.

of the steam in the valve-box causes them to spring and so approach one another, making the combined valve tight on each face. At the same time, the pressure is not so much as to cause very great resistance to the motion of the valve. This form of valve has been found in practice to work very well indeed, and many months of successful use have produced little or no wear on the valve faces.

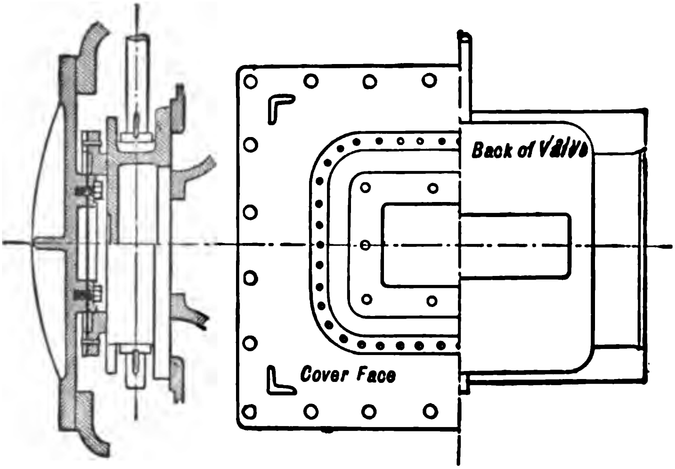

Fig. 91.—Dawe & Holt's Patent Relief Frame.

Fig. 90 shows a piston-valve adapted by Mr. Thom with the same principle as the Trick valve.

This arrangement rather complicates the piston valve and renders it far less simple than in the ordinary form, but there are no practical difficulties in the way of accomplishing what Mr. Thom aims at—viz., to admit some steam for the purpose of cushioning at one end of the cylinder from the exhaust just commencing at the other end. This, when quiet running is necessary, it is hardly needful to say, is of considerable advantage with the low-pressure cylinder, but is not so necessary to the medium-, and not at all necessary to the high-, pressure cylinder of a triple-expansion engine.

Dawe & Holt's Patent.—This consists of a rectangular cast-iron frame, fitting steam-tight to the face on the back of the valve, and riveted to a steel or bronze sheet, which is itself secured steam-tight to the valve (fig. 91). The diaphragm is of hard copper or phosphor bronze, and only 18 B.W.G. thick, and so allows the frame to move slightly to suit the valve. This is so made that the frame is pressed back about $\frac{1}{16}$ inch when the valve is in place, and the frame itself, being

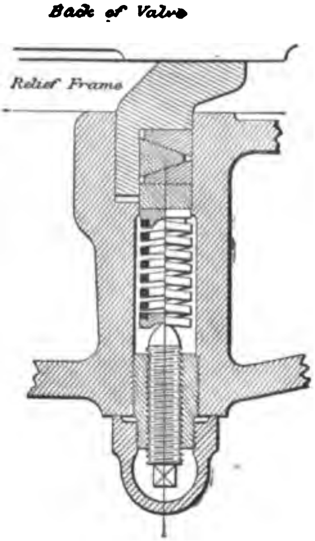

Fig. 92.—Spiral-spring Packing to Relief Frame.

exposed to steam pressure, is always pressed against the valve, the area enclosed by the frame being the amount of relief given to the valve.

Common Relief Frame.—The ordinary method of relieving the back of the slide valve from steam pressure is by means of a frame of circular or rectangular form, fitting steam-tight, but freely, into a recess in the valve-box cover, and pressed against the smooth face at the back of the valve by set screws, an elastic material, such as india-rubber, or spiral metallic springs as shown in fig. 92, being interposed between the screws and the frame, in order to give greater freedom to it.

Fig. 93 is a modification of the common plan, and is especially applicable to valves of nearly square form. In this case the relief frame is circular, fitted into a recess in the back of the valve, and pressed out by a star-shaped steel spring.

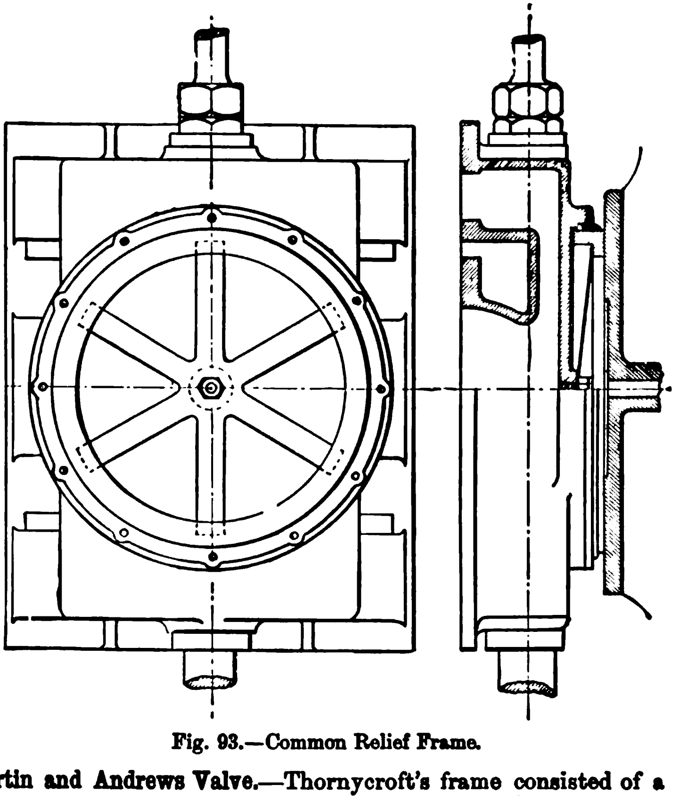

Fig. 93.—Common Relief Frame.

Martin and Andrews Valve.—Thornycroft's frame consisted of a casting with recesses and faces corresponding to the ports and bars of the cylinder face; the valve was with flats and recesses on its back corresponding to its face. The frame rested on legs touching the cylinder face, and just touched the valve back so as to keep the steam off the surfaces of contact. Figs. 94 and 94*a* represent a modification of this plan so improved that what appears as an ordinary single-ported loco valve really acts as a double-ported one by allowing steam to enter at both front and back edge, and to pass to exhaust at two places in the same fashion—viz., by way of the little port crossing from front to back of the valve at each end. This invention is patented by Messrs. Martin and Andrews, and much used by the former engineer for the cylinders of the large and successful quadruple engines turned out by him from the Flushing works.

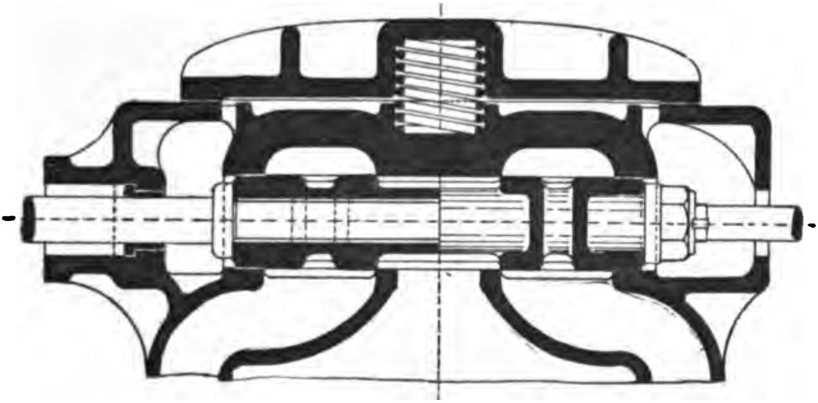

Fig. 94.—Martin and Andrews Valve Relief.

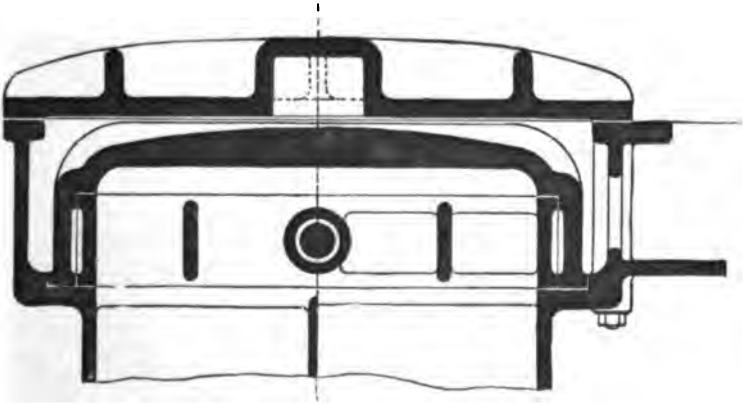

Fig. 94a.—Transverse Section of Martin and Andrews Valve.

There are many other plans for preventing slide-valves from pressing unduly on the cylinder face, some by means of springs, and some by pistons acting perpendicularly to the direction of motion of the valves, but none of them are free from objection, nor have any given greater satisfaction than Church's (fig. 95) very ingenious arrangement.

Through Exhaust Valve.—Fig. 96 shows a low-pressure double-ported valve arranged to exhaust through its back direct; provided the packing-ring is well designed, well made, and kept in good order, this is a safe and useful one, as it permits of the face being much closer to the cylinder than ordinarily, and a perfectly unobstructed lead to the condenser.

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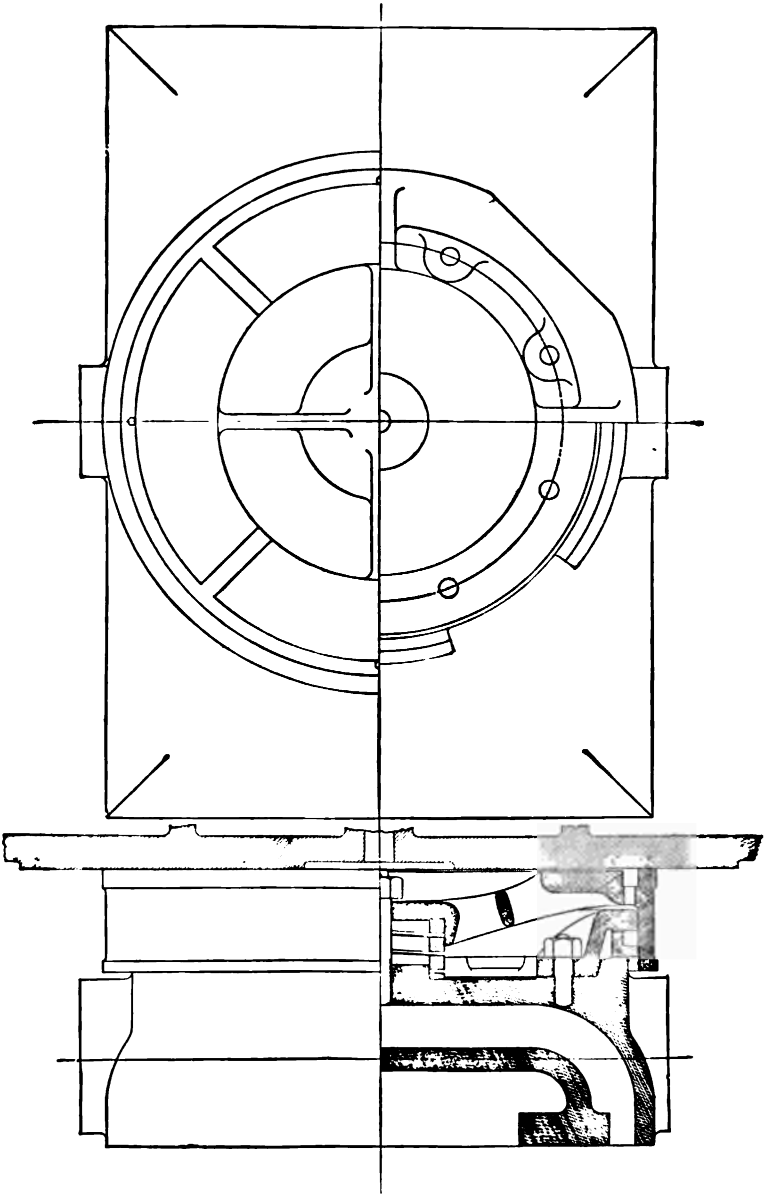


Side Elevation. | **Section through Middle of Valve.**

Fig. 95.—Church's Patent Relief Frame.

A. Plan of Relief Ring. | **B. Plan with Ring Removed.**

Back Guides and Springs.—To prevent slide-valves from being forced from the cylinder face, it is necessary to have guides behind the valve, with suitable rubbing surface on the valve itself; but as it is not always easy to provide such guides, and since, when the valve and face have worn somewhat, they fail to keep the valve to its face, it is generally preferable to fit a pair of flat bar springs, whose backs press on rubbing strips on the valve,

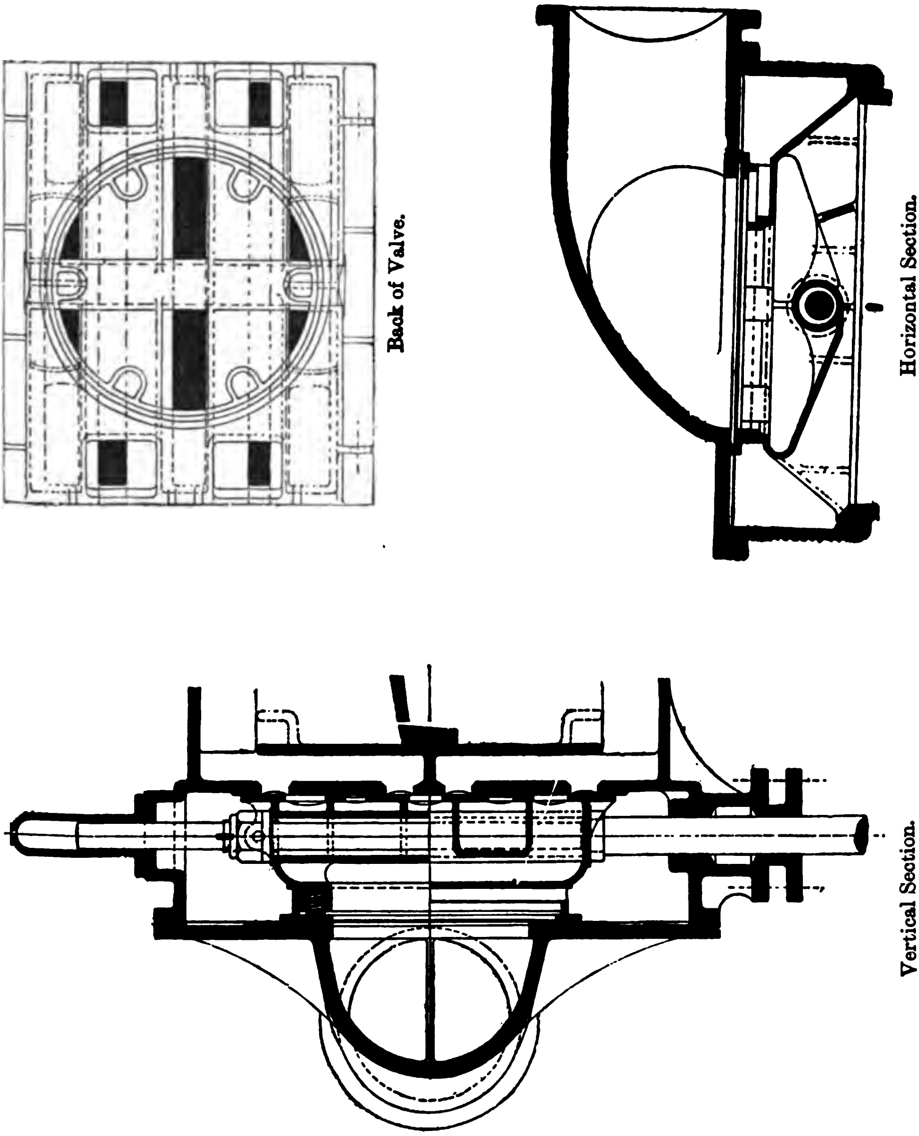


Fig. 96.—Valve with Exhaust through the Back.

and the ends on a fixed part, one end of each being secured, and the other free to slide on the flattening of the arc.

Balance Pistons.—Since the weight of the valves, their rods, and gearing, in the case of vertical engines, is taken by the eccentric straps, and if it were unbalanced the top half of the straps would be subject to more wear than the lower, it is advisable to provide some means of avoiding this. The readiest is by fitting to the top end of the valve-spindle a piston (figs. 29 and 88), which works in a cylinder, provided in the valve-box cover or top; this piston is of sufficient area that the steam-pressure on it affords the balance required, and it also acts the part of a guide for the spindle. When the eccentric-rods are very long, it is better to give an excess of area to the balance piston, so that the rods may be *always in tension*, instead of alternately in tension and compression. Since the pressure in the valve-box of the low- and medium-pressure cylinder of a compound engine is constantly varying, it is better, especially when the above object is aimed at, to place the balancing cylinder outside the box, and supply steam from the boiler to the under side of the piston, so that the balancing force may be constant; but, on the other hand, since the resistance of the valve varies with the pressure in the valve-box, there is not the same necessity for so much balancing force as would be the case were it otherwise; so that, on the whole, the piston exposed to the pressure of the valve-box is sufficient for the needs of most engines.

The top of the balance cylinder of the high- and medium-pressure valves should be connected to the next receiver by a small pipe, and that of the low-pressure valve to the condenser.

Joy's Assistant Cylinder.—In the hands of Mr. David Joy the balance piston with its cylinder has been transformed by a very ingenious and simple device, from merely passively supporting the weight of the valve and its gear, to actively aid the link motion in moving the valve up and down. Fig. 97 shows the method by which this was accomplished at the outset; here the steam is admitted to the top and bottom of the piston through passages in the piston itself which open to steam ports in the sides of the cylinder at or near the end of the stroke. The exhaust is also through ports in the side of the cylinder which are opened and closed by the piston itself towards the end of the stroke. Fig. 97*a* shows another method, which Mr. Joy calls the compound system, inasmuch as the steam contained in the body of the piston, after partially expanding on the upstroke, is passed by means of a recess near the top of the cylinder into the space above the piston so as to expand there and so impel the piston downward. The steam obtained access from the underside to the interior of the piston by means of another recess near the bottom. The valve spindle rod is reduced in diameter for some little distance below the piston, and, passing through a sleeve with port holes in it, does duty as a valve for the admission and cut-off of steam. Just before cut-off the piston passes the lower recess so as to allow its interior to fill. The piston face in way of the exhaust ports is, of course, plain, so as to completely cover them and prevent the steam inside the piston from passing to exhaust.

Fig. 97*b* shows a set of indicator diagrams taken from an assistant cylinder working under these conditions, and they certainly are remarkable, considering the simple method by which the steam is distributed and released. These assistant cylinders have been fitted by several firms to fast-running engines which have of necessity heavy slide-valves, especially those of the low-pressure cylinder. The results have been very satisfactory, and the headgoing, eccentrics, &c., relieved to a very considerable extent from the heavy work entailed in operating such valves at a high speed.

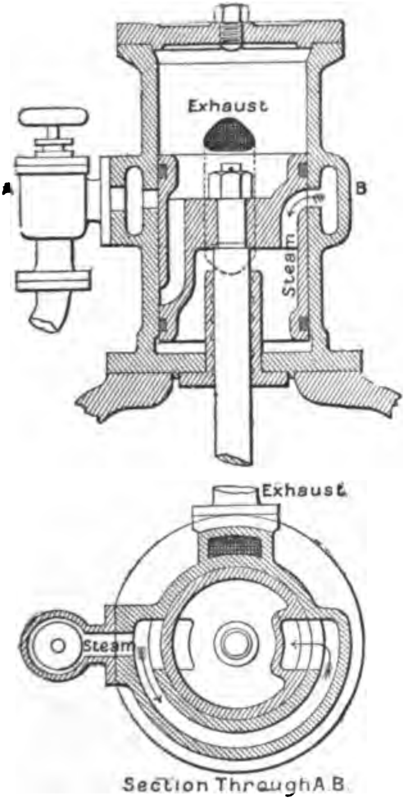


Fig. 97.—Joy's Assistant Cylinder.

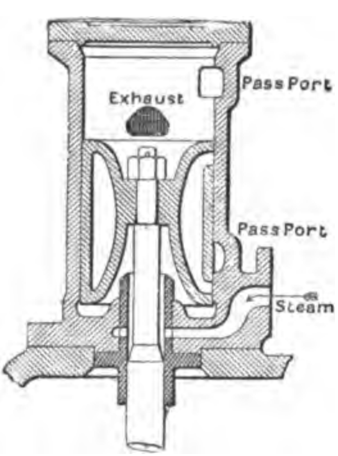


Fig. 97*a*.—Joy's Improved Assistant Cylinder.

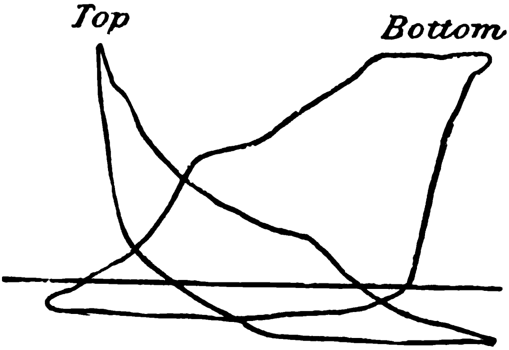


Fig. 97*b*.—Indicator Diagram from Joy's Assistant Cylinder (fig. 97*a*).

Valve-rods or Spindles.—Since it is possible for a slide-valve to be exposed to the pressure of steam on its whole area, without any relief due to the ports, &c., and this may occur even when the valve is fitted with relief frames, it is better to assume this in making all calculations for determining the sizes of the parts to move it.

If L be the length of a valve, and B the breadth in inches, p the maximum absolute pressure to which it is exposed in pounds per square inch, then,

$$\text{Maximum pressure on the valve} = L \times B \times p \text{ lbs.}$$

The coefficient of friction should be taken at 0·2, or that of metallic surfaces rubbing together dry, as this is the worst condition likely to occur, then,

$$\text{Load on valve-rod} = 0{\cdot}2\,(L \times B \times p)\ \text{lbs.}$$

Since the spindle has to take stresses similar to those on a piston-rod, it may be dealt with in the same way.

A stress of 3000 lbs. per square inch should be allowed in the case of comparatively long rods, and 3600 when very short and well-guided. Hence

$$\text{Diameter of slide-valve rod} = \sqrt{\frac{L \times B \times p}{F}}.$$

When the rod is long F = 12,000.
„ „ short F = 14,500.

In the case of the valve of the low-pressure cylinder, the pressure should be taken at 25 lbs. absolute, as the pressure in the receiver is seldom as high as 15 lbs. above that of the atmosphere. For convenience, the rods for both high-, medium-, and low-pressure cylinders should be alike, and consequently the size should be the largest given by calculation from the above rules.

When the guides of the valve-rod are above the rod-end (*v.* fig. 98), so that in reversing the link a severe bending strain would come on the rod, it should be somewhat larger in diameter in the guide. In cases of this kind, F should be taken at 20 per cent. less than given by the above rules for that part.

Valve-rod Bolts.—When the joint at the valve-rod end is made capable of adjustment by means of two bolts (fig. 103); then,

$$\text{Diameter at bottom of thread} = \sqrt{\frac{L \times B \times p}{H}}.$$

When the bolts are of iron, H = 33,000; when of steel, 38,000. This same rule applies to the bolts of eccentric-rod ends, and to the bolts at the butt-end of eccentric-rods.

For the bolts of eccentric straps, H = 28,000 for iron, and 32,000 for steel.

Valve-rod Guides.—The valve-rod end should always have a guide, so that the side strain due to the obliquity of the eccentric-rods, or to the action of the link-motion, shall not only not bend the rod, but shall not come on the gland and so cause leakage, &c., through the stuffing-box. The kind of guide depends very much on the link-motion used, as what suits one form is not applicable when another kind is used. The simplest form consists of a cast-iron bracket with an eye bushed with brass, through which the rod passes; but since the rod soon wears away the bush, if subject to much side strain, this is not a satisfactory plan for large engines, although a convenient and fairly trustworthy one for small ones. If the rod is made of larger diameter in wake of the guide, and planed flat on the sides which are exposed to pressure, and the bracket fitted with brass liners, shaped like a common key, so as to be secured by screws through the gib ends, a simple and efficient means is provided for taking all side-pressure, but the drip from the gland is apt to wash out the oil. A better but more costly one is to fit a bracket to the valve-rod above the link, having a shoe like that of a piston-rod, and running on a guide like that of a piston-rod end. In large engines it is fitted with a slipper, &c., for adjustment (fig. 98).

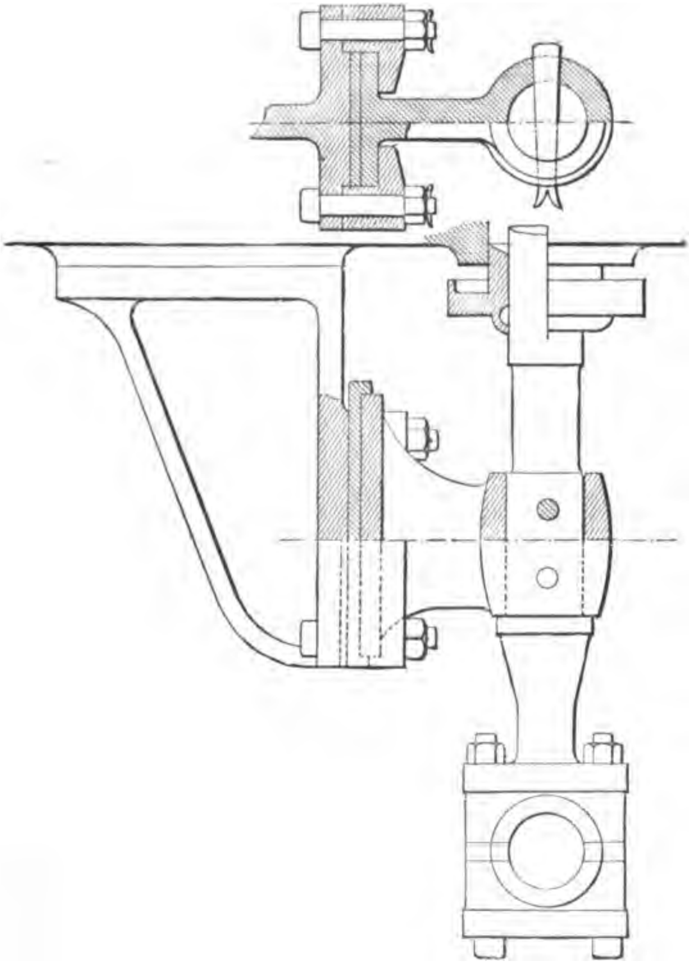


Fig. 98.—Valve-rod Guide.

Slot links require a different kind of guide when they work on overhung pins, which is often the case, of necessity, in horizontal engines, to obtain eccentric-rods of sufficient length. Then the valve-rod is fitted into the socket of a guide-piece, shaped somewhat like a bayonet, having a shank of square section and working in a guide bracket fitted to the framing (*vide* fig. 15*a*).

Proportions of Slide-valves are found as follows :—

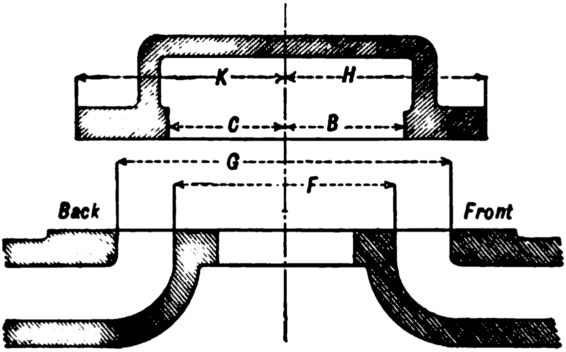


Fig. 99.—Proportions of a Common Valve.

Let x be the *outside* lap of the valve at the front end, and y that at the back end. Then,

$$H = \frac{G}{2} + x; \qquad \text{and } K = \frac{G}{2} + y.$$

Let z be the *inside* lap at the front, and w that at the back. Then

$$B = \frac{F}{2} - z; \qquad \text{and } C = \frac{F}{2} - w.$$

To keep the valve as small as possible, E need be only a little wider than the steam ports, provided that the half travel does not materially exceed the width of bar.

The Double-ported Valve dimensions follow similar rules to the above.

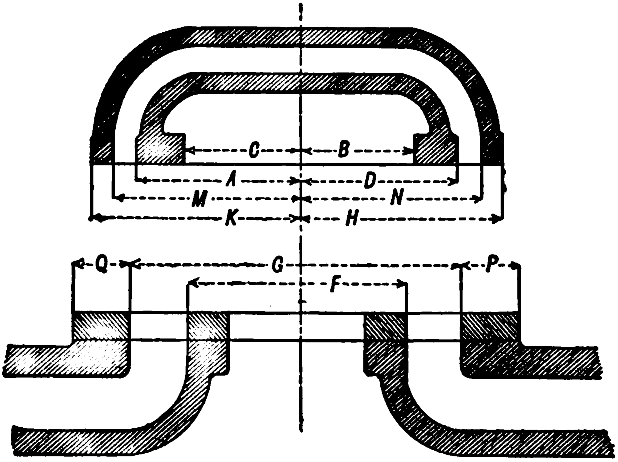


Fig. 100.—Proportions of a Trick Valve.

Let x, y, z, and w be the laps as before. Then

$$H = \frac{G}{2} + x; \qquad \text{and } K = \frac{G}{2} + y.$$

Also,

$$B = \frac{F}{2} - z; \qquad \text{and } C = \frac{F}{2} - w.$$

$$A = \frac{G}{2} + \tfrac{1}{8} \text{ inch}; \qquad \text{and } D = \frac{G}{2} + \tfrac{1}{8} \text{ inch}.$$

The openings through the valve laps or covers must be as large as possible, but need not exceed the ordinary opening of the valve to steam at the outer edge; then

$$G + P = K + N;$$

and

$$G + Q = H + M.$$

Valve Gear: Link Motion.—The common form of motion employed to work the valves of a screw engine consists essentially of two eccentrics keyed on the crank-shaft in such a position relatively to the crank that when one is operating on the valve, the engine will propel the ship ahead, and is said to be in *head-gear*, and when the other, the engine will propel the ship stern first, and is said to be in *stern-gear*, their rods being connected by a bar or bars on which is a block to which the valve spindle is attached. This bar connection is called the link, and is of such a form that by sliding it through the block, the *head* or *stern* eccentric may at pleasure be brought to operate on the valve.

In designing a link gear, the most important objects are to give the valve such motions as shall cause it to open to steam slightly before the piston is at the end of its stroke, the amount by which it is open at the end of the stroke, or commencement of the next stroke, being called the *lead* of the valve; to open then fully, and close at the required period of the stroke of the piston called *cut-off;* to confine the steam during the remaining portion of the stroke, so as to *expand* in the cylinder, and at or near the end of the stroke to allow the steam to escape from the cylinder, called *exhaust;* to close the port again before the end of the stroke, so that the piston *compresses* the steam remaining in the cylinder and port. These operations should be effected with the expenditure of as little power as possible, and with this end in view the motion of the link should be, as far as can be, limited to moving the valve only; consequently the link itself should have no sliding motion longitudinally, called *slotting motion*, in the block, but only the angular motion due to the two eccentrics. A perfect valve motion is such that the valve opens to steam *wide* immediately the crank has passed the *dead centre*, and remains open during the admission of steam, so that there is no wire-drawing; the valve closes suddenly, and remains closed during expansion; at the end of the stroke it opens wide to exhaust, and remains in that state during the greater part of the period of exhaust, when it closes suddenly, and remains closed till opening again to steam. This is not obtainable with the ordinary link motion, nor to its full extent with any motion when one valve only is employed for both ends of the cylinder, because the period of cut-off at one end does not, as a rule, correspond to the period of compression at the other end; but there are valve gears which have two periods of very quick motion, and two of very slow in each cycle, which very closely fulfil the above conditions, and which will be noted later on.

Slot Link.—This, which is one of the oldest forms of link, is still retained by many engineers, and is well adapted to the circumstances of several forms of engine, such as the oscillating paddle-wheel engine, and all engines in which there is not direct connection between the eccentric-rods and the valve-rod, and also in some of the horizontal engines when it is either impossible or inconvenient to have direct connection.

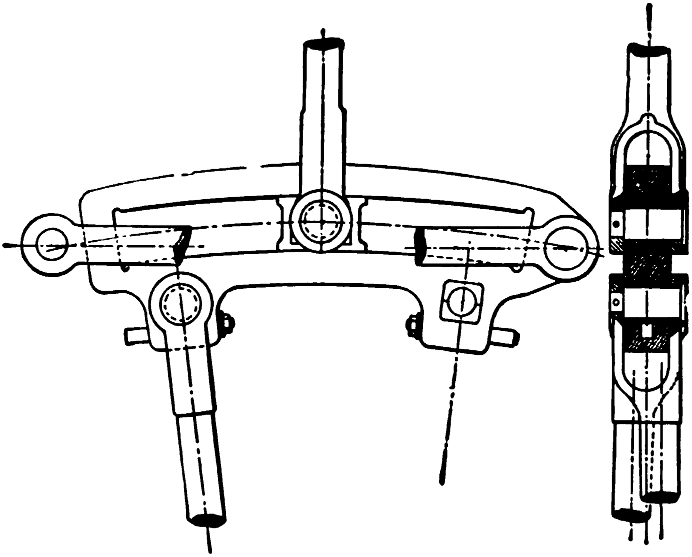


Fig. 101.—The Slot Link.

Fig. 101 is an illustration of the ordinary slot link, having adjustment for the sliding block and eccentric-pin brasses. Locomotive engineers prefer, as a rule, to have the pin-holes fitted with hard bushes rather than adjustable brasses; but this opinion is not shared by marine engineers, and chiefly on the ground that in a foreign port it is seldom possible and never convenient to engage the services of workmen and tools to renew these bushes when so badly worn as to require renewal.

This kind of link is generally suspended from the end next the head-going eccentric-rod, at a point in line with the arc through the block-pin; and if the pin in the lever, which operates on the link to reverse it, is placed in the proper position, there is very little slotting motion indeed when working in *head*-gear. The same remark applies to the position of the pin when in *stern*-gear, except that the amount of slotting motion is somewhat greater of necessity; but since a marine engine, as a rule, works very little in *stern*-gear, and its efficiency there is of small consideration comparatively, this defect is of little moment. Locomotive engineers still keep to the slot link, as it has no small bolts and nuts to rattle loose and give trouble, as might be the case with other gears; but in the locomotive the eccentric-rod ends are connected to the ends of link at the place where the drag-rod is in fig. 101.

Position of Suspension Pin.—To obtain the best position of lever pin, it is necessary to draw out the path of the centre of pin in the link end through one revolution of the engine when the link has *no slotting motion*. The path so found is like an attenuated figure 8 in head-gear, and somewhat more pronounced in stern-gear. The arc of a circle of radius equal to the length of the suspension or bridle-rods, is then drawn through each of these figures in such a way that there is the least possible deviation of the figure on either side; that is, the arc is the centre line of the figure, if the deviation on one side equals that on the other. The centres of the circles to which these arcs belong should be the centres of suspension of the bridle-rods, or position of pin in reversing lever end. By drawing arcs of circles of radius equal to the length of the reversing lever from these two centres, the points of intersection are the two possible positions for the centre of weigh-shaft.

This same method of construction is suitable to all kinds of links, and for all positions of the point of suspension of the link.

When it is necessary that the motion shall be as efficient in *head*-gear as in *stern*-gear, the link should be suspended from a point in the line dividing it symmetrically, and by preference at the intersection of this line with the arc through the centre of block-pin, so that the centre of suspension is in line with the centre of block-pin when in *mid-gear*. When this is so, the pins for suspending the link are on side plates bolted to the sides of the link.

The distance from centre to centre of eccentric-rod pins should not be less than two and a half times the *throw* of the eccentrics, and is usually, when space permits, two and three-quarters to three times. The *throw* of the eccentrics in this case is, of course, equal to the travel of the valve when in *full* gear.

Size of Slot Link.—Let D be the diameter of the valve spindle, R being the revolutions per minute, and F = (13,500 − 10 R), from the following calculation

$$D = \sqrt{\frac{L \times B \times p}{F}};$$

then

Diameter of block-pin when overhung - - - - = D.
„ „ „ secured at both ends - = 0·75 × D.
„ eccentric-rod pins - - - - - - = 0·7 × D.
„ suspension-rod pins - - - - - = 0·55 × D.
„ „ rod pin when overhung - = 0·75 × D.
Breadth of link - - - - - - - - - - = 0·8 to 0·9 × D.
Length of block - - - - - - - - - - = 1·8 to 1·6 × D.
Thickness of bars of link at middle - - - - - = 0·7 × D.
If a single suspension-rod of round section, its diameter = 0·7 × D.
If two suspension-rods of round section, their diameter = 0·55 + D.

The objections to the slot link are, that it is an expensive one to make, and that, owing to the eccentric pins and the block-pins being out of line, there is always an uneasy motion about the block-pin, and more slotting motion of the block. The former objection is valid, especially when the link is made of wrought iron, but when the link is made of cast steel it is not so expensive as some other forms. The uneasy motion is often due to bad design, for when well designed and carefully hung, it will work very satisfactorily.

Single-bar Link.—This kind of link consists of a single solid bar, of rectangular section generally, and having the eccentric-rods connected to each end, and a sliding block between, to which the valve spindle is connected. The form of link, although tried by more than one eminent firm of engineers, has gradually been dropped, except by the firm that introduced it.

Double-bar Links.—There are two kinds of double-bar links; one (fig. 102) having the eccentric-rod ends, as well as the valve-spindle end between the bars, so that the travel of the valve is less than the throw of the eccentrics; the other (fig. 103) has the eccentric-rods formed with fork ends, so as to connect to studs on the *outside* of the bars, and thus admits of the block sliding to the end of the link, so that the centres of the eccentric-rod ends and the block-pin are in line when in full gear.

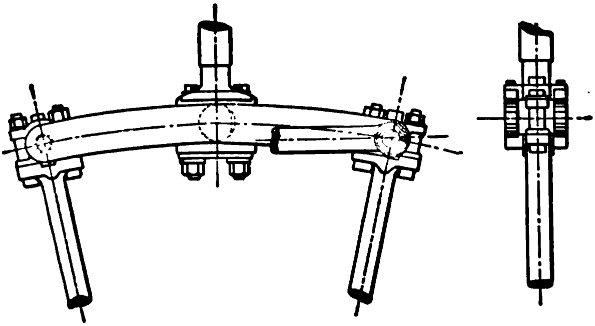


Fig. 102.—Double-bar Link with Rods inside.

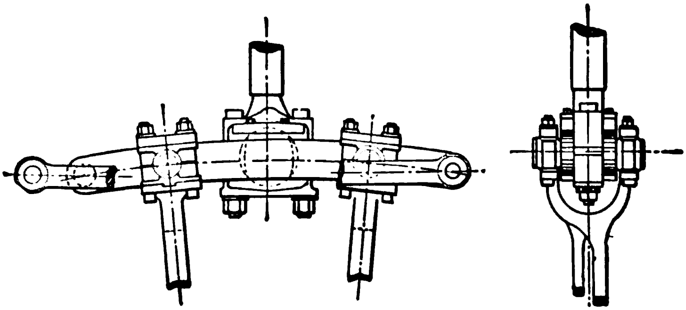


Fig. 103.—Double-bar Link with Rods outside.

The former plan is cheaper to make, is simpler in construction, and has fewer parts to get out of order and adjust; and when adjustment is required, it is easier to make, and there is less chance of its being done improperly. When in *head*-gear, part of the work of moving the valve is done by the *stern*-going eccentric, so that the wear is not limited to the one eccentric strap. When properly hung, the slotting motion is exceedingly small, and the valve motion is as perfect as with the other form. The objection to it is that the eccentrics are larger in diameter than those with the other links, and the links themselves are longer, and more space is required for the eccentric-rods to move in.

Size of Bar Links.—Let D be the diameter of valve spindle, found as before.

Fig. 102, distance between centres of eccentric pins, 3 to 4 times throw of eccentrics.

Depth of bars . . . . = 1·25 × D + ¾ inch.
Thickness of bars . . . = 0·65 × D + ¼ inch.
Length of sliding block . . = 2·5 to 3 × D.
Diameter of eccentric-rod pins . = 0·8 × D + ¼ inch.
" centre of sliding black = 1·3 × D.

Fig. 103, distance between eccentric-rod pins 2½ to 2¾ times throw of eccentrics.

Depth of bars . . . . = 1·25 × D + ½ inch.
Thickness of bars . . . = 0·65 × D + ¼ inch.
Length of sliding block . . = 2·5 to 3 × D.
Diameter of eccentric-rod pins . = 0·75 × D.
Length " " . = 0·80 × D + ¼ inch.

Diameter of eccentric bolts (top end), at bottom of thread = 0·42 × D when of iron; and 0·38 × D when of steel.

These bars should be of a very good description of iron and case-hardened, or of hard steel; the latter is of course free from seaminess and stronger. The eccentric-rod pins of the kind of link (fig. 103) are usually forged solid with the bars, but there is no absolute need of this, and it adds very much to the cost, both of manufacture and renewal when worn. Since the wear on these pins is limited to a very small portion of their circumference, it is not unusual to file away the parts which are not subject to wear, so as to admit of the brasses being closed when worn. When loose pins are fitted, they should be steel, and hardened, so that all wear may come on the brasses which are capable of adjustment.

In another arrangement of bar-link motion, the sliding block is divided, and on the outside, while the eccentric-rod ends are between the bars. This, while having some slight advantages, is on the whole very clumsy, and the block-pins wear badly; besides which, the link can be only suspended from the extreme end. Mr. Martin, of Flushing, adopts an ingenious combination of these bar links for the large engines he turns out. He fits three eccentrics to each valve; the rods of the two outer ones are connected each to a pin on the outside of the link, as in fig. 103, and act in "ahead" gear. The rod of the centre eccentric is connected to a pin between the bars, as in fig. 102, and acts in "astern" gear.

Single Eccentric Gear.—The valves of slow-working engines can be worked by a single eccentric, which is free to move round on the shaft from the position for *head*-gear to that for *stern*-gear; it is driven by a key or *stop* fixed to the shaft, pressing against a shoulder on the side of the eccentric. The eccentric sheave is balanced so that it will stop in any position, and only move when driven by the *stop* on the shaft. The eccentric-rod is so fitted that it may be disconnected from the valve-rod or its gear. This is generally effected by providing a *gab* or gap in the eccentric-rod end instead of a pin-hole, which allows the eccentric-rod to be lifted from its pin by suitable gearing. When the eccentric-rod is disconnected from the valve gear, the valve ceases to move, and the engine comes to rest. To restart the engine the valve must be worked by hand until the gab can be brought in line with the pin; if the motion of the engine is reversed the shaft will move around, the eccentric remaining motionless until the stop on the shaft comes in contact with the other side of the projection on the sheave; it is then in the right position for driving the valves, and the eccentric-rod may now be dropped into gear. This method admitted of the paddle engine being handled very dexterously when the valves were not so large and the steam pressure so high as to prevent their being moved by hand, and was the one generally adopted in engines of moderate power long after link motion was invented and in general use on locomotives.

There have been many other methods of moving a single eccentric from ahead to astern position; some of which by sliding wedges were at one time received favourably, but are now almost forgotten.

Hackworth's Dynamic Valve Gear.—The motion of a point on a rod, one end of which moves in a circle, and the other on a straight line passing through the centre of that circle, is an ellipse whose major axis coincides with the straight line. If, however, the end of the rod slides on a line inclined to this centre line, the major axis of the ellipse will be inclined.

Mr. J. W. Hackworth's valve motion works on this principle, for his gear has a single eccentric keyed to the shaft *exactly opposite to the crank;* the eccentric-rod has its end attached to a block, which slides on a guide-bar inclined to the line through the centre of shaft; and a rod from about the middle of the eccentric-rod is connected to the valve-rod, which works in a line at right angles to the line through the eccentric-rod when in mean position. If the guide-bar on which the eccentric-rod block slides be moved so as to reverse its inclination, the inclination of the axis of the ellipse is reversed, and the motion of the engine reversed. Here the *motion* of the valve is due to the eccentric, and the *lead* to the inclination of the slide-bar.

The great advantage of this gear, beyond the saving of an eccentric, is the better motion imparted to the valve, inasmuch as there are two quick and two slow motions in a revolution; the quick ones occurring at opening and cut-off, and the slow ones during entry and at exhaust previous to opening. A large variation in the amount of cut-off is possible with this arrangement, without wire-drawing from small opening and slow closing of the port, as is the case with the common link motion. The chief objections urged against this gear are the excessive friction, and consequent wear on the sliding blocks, and the liability of so many pins to derangement. The first of these is the most valid, and it has been overcome by fitting rollers instead of sliding blocks. In both ways, however, this gear has worked fairly well; and for engines of small power it is a very convenient arrangement, especially when much variation in cut-off is required.

Since the valve-rods with this gear do not come in line with the piston-rods, or immediately over the line of shafting, the valves and their chests are removed from the usual positions, so as to admit of the cylinders being closer together; the engine is consequently shorter, cheaper, and occupies less space. This suited the expansive screw engine and paddle engines of the diagonal type admirably; for the two valves of an expansive engine being in one common chest, could be examined through a door in front, of ample dimensions to admit a man; for the compound engine the valve-boxes are side by side with short-pipe connections (*vide* fig. 8); but generally it allows of little space for the receiver, and the exhaust from the low-pressure cylinder of screw engines must pass through a belt within the receiver, or else this belt has to traverse more than half the circumference of the cylinder.

Marshall's Valve Gear is a modification of Hackworth's, and differs from

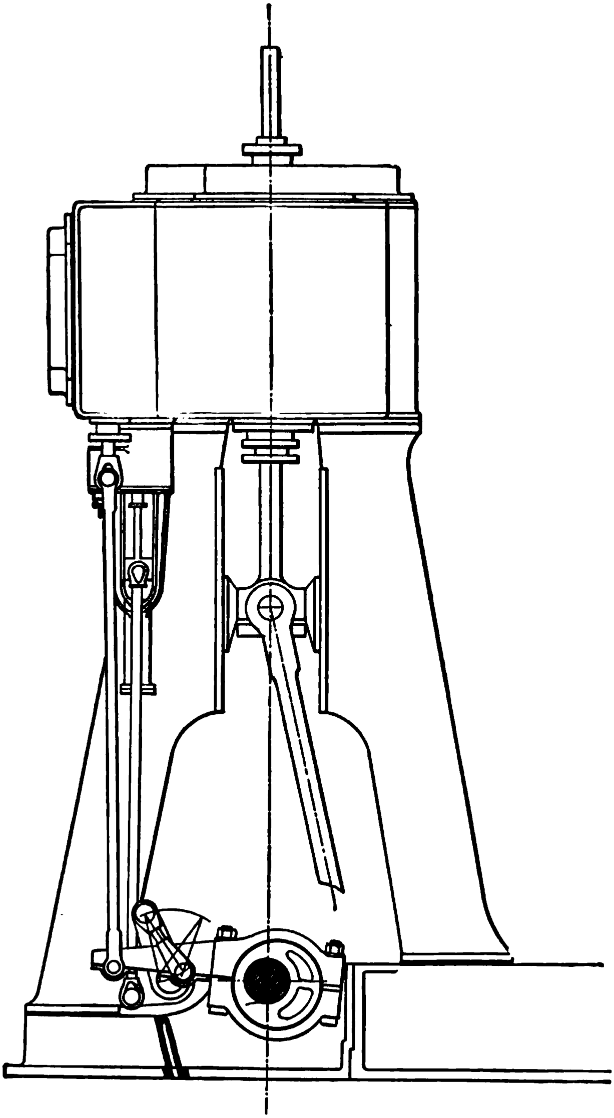


Fig. 104.—Marshall's Patent Valve Gear.

it in the method of getting the oblique motion of the rod end. Fig. 104 shows the plan adopted by Mr. F. C. Marshall, and also illustrates generally what has been said of Hackworth's. Here the eccentric-rod is hung, by means of a rod from the end of a lever on a reversing shaft, in such a way that it moves on the arc of a circle inclined to the centre line. The motion is not quite so perfect as with the inclined sliding bar, and necessitates double ports to the bottom end of the slide-valve, in order to get as much opening to steam there as at the top end; but there is less friction, and, on the whole, it works most satisfactorily. The pins require to be of good size, and they should all have adjustable brasses to provide for the large amount of wear which of necessity comes on them.

Joy's Valve Gear.—Fig. 105 shows the ingenious arrangement whereby Mr. David Joy has avoided altogether the use of eccentrics. He obtains the motion from the connecting-rod, and qualifies it by a sliding block like Hackworth, or a suspension-rod like Marshall. The motion thus obtained is a very perfect one for a slide-valve, as the two quick and two slow periods are just when required, the amount of opening is equal at both ends of the valve, and early cuts-off can be effected without excessive leads and compressions or premature exhaustings. This gear has been applied with great success to locomotives, where the saving of space for the eccentrics admits of longer crank-pins and journals; it has also been taken up by a few marine engineers. The chief objections to this gear are that it comes in the way of the principal working parts, and makes them a little difficult to get at when working, and also that a small amount of wear on the joints of the gear would cause a defect on the valve motion, and produce a serious amount of rattle of the gear. These, however, can be got over by making the pins substantial, and all the joints adjustable.

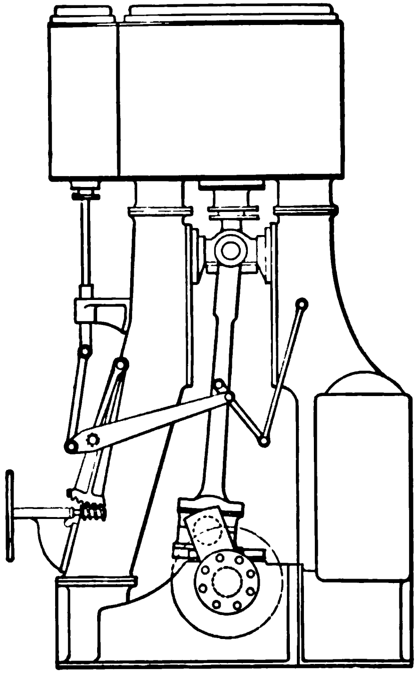


Fig. 105.—Joy's Valve Gear.

Sell's Valve Gear.—Here the valves are worked by an independent shaft, which derives its motion from the main shaft by means of wheel-gearing; between these two shafts are two intermediate wheels gearing into one another—one of them gears into the wheel on the main shaft, and the other into that on the valve shaft. These two intermediate wheels are carried on a frame which can be lifted up and down, and is guided in such a way as to keep the wheels always in proper gear with their corresponding wheels. If the gearing is set so that the valves are moved by their crank-shaft, and drive the engine *ahead* when the frame is at the bottom of its traverse, the moving of the frame to the top of its traverse so alters the relative angular position of the shafts, that the valves will be set to drive the engine astern.

This is easily understood by supposing the engine at rest, and the frame raised; the wheel geared into the one on the crank-shaft, in moving round the shaft also turns on its own axis, and in so doing turns its companion-wheel, which turns the wheel on the valve shaft, and consequently the valve shaft itself.

This gear was fitted to many engines in the English and foreign navies, and was especially suitable for the three-cylinder horizontal engines for which it was originally designed. It had, however, the objectionable feature of wheel-gearing, which, when new, made a great noise, and was liable to accident from *racing* of the engines, but when worn, these are magnified very considerably by the *back lash* which results. Had the wheels been machine-cut accurately to shape and of sufficient size, this gear might have been in use to-day, as it was a very convenient one for horizontal engines, as also for large vertical ones. A similar arrangement has been used by certain engineers for vertical triple engines, but the auxiliary or valve shaft was driven by an auxiliary engine, which had a sensitive-ball governor to regulate it. In rough weather the wheels connecting the two shafts were thrown out of gear, and the main engine was quite governed by the small one.

Expansion Valves.—When ordinary link motion is employed, and an earlier cut-off than half-stroke is required as a normal condition, it is necessary to effect it by means of an independent valve, usually called the expansion valve. These valves are generally common plates, sliding either on a face with ports on the side of the valve-box, or on a face on the back of the ordinary slide-valve. In the latter case there are two methods, commonly known as *the inside cut-off* and *the outside cut-off.*

Gridiron Expansion Valves.—When the valve worked on an independent face on the side of the valve-box, it consisted of a plain plate with steam ports in it, corresponding to the steam ports on the face; there was sufficient lap, and the gear was so set that it remained closed after it had cut-off, until after the main valve was closed to the cylinder, when it might open again so as to fill the valve-box, and be ready to supply and cut off steam from the other end of the cylinder. The variation in cut-off is effected by varying the travel, and the equalisation of cut-off at each end is effected by varying the proportion of lap on each side of the ports. The variation in travel is obtained by means of a link on which the block to which the expansion eccentric-rod is attached slides, so as to vary the length of lever; the larger the travel of the expansion valve, the quicker is the steam cut off, and the smaller the travel, the later is the cut-off, until, finally, the travel is so small, that the expansion valve does not close at all. The variation in the lap of the expansion valve is effected by means of the curvature of the link.

Some engineers preferred the mean position of the expansion valve to be with the ports closed; in this case the quickest cut-off is effected with the least travel, and consequently with least opening; but since the piston speed is comparatively slow when receiving steam with the early cut-off, a reduction of opening will not be felt; whereas the slow cutting-off when carrying steam to mid-stroke, effected by the valve, when the late cut-off is with least travel, produces considerable wire-drawing.

If the engine is generally to work with a cut-off at from $\frac{1}{4}$ stroke to $\frac{1}{2}$ stroke, then the former valve arrangement is the better; if the cut-off is to be, as a rule, from $\frac{5}{8}$ to $\frac{3}{8}$ the stroke, then the latter is better, except that it is somewhat inconvenient to have the ports closed when the valve has least travel, and also that when the expansion valve is not required it must be moving at its highest speed.

The chief objection to this kind of expansion valve is the large amount of clearance space between it and the piston, for the steam expands in the valve-box, as well as in the cylinder, until the main valve closes. It was, however, very generally used in the Navy for the cut-off for cruising speeds, and had the advantage of not interfering with or in any way affecting the main valves in case of anything happening to it.

Outside Cut-off Valves.—When the expansion valve works on a face on the back of the main valve, the latter is provided with steamways through it from the ports on its back to the ports on its face. If the main valve is single-ported, it is like a common locomotive slide-valve with enclosing ends, so that steam passes to the cylinder only through the passages thus formed. There are then two ports on the back of the valve, one of which is covered at cut-off by the expansion valve, and remains covered until the main valve cuts off. The expansion valve may consist of a single flat plate, which, when in mean position with respect to the main valve, is between the ports on the back of the main valve. If any variation of the cut-off is required in this case, it is effected by varying the travel of the expansion valve with respect to the main valve, or by altering the *lead* of the expansion eccentric. The latter plan would generally be very inconvenient, and, consequently, when such a valve is fitted, the variation is effected by varying the travel.

On account of the cut-off being effected by the outside edges of the expansion valve and the ports in the main valve, these are called outside cut-off valves. Since, however, the motion of the valve will be slow when cutting off with a small relative travel, it is unusual to fit a single plate, but to divide it and arrange the gear so that the two plates may be moved apart from one another so as to virtually increase their lap, and cause them to cut off earlier with the same travel. The result of this arrangement is that the expansion valve moves always at the same velocity and through the same space; the cut-off occurs when moving at or near the maximum velocity, whatever be the period of cut-off, and there is consequently little or no *wire-drawing* at the ports. The plates should be so designed that when cutting off at the earliest required period, they do not overrun the port so as to pass steam through at the *inner* edges, and that they may approach close enough together as to allow of a late cut-off if required.

The usual method of altering the position of the plates is by securing them by nuts to a common spindle, on which is cut a right-handed thread for the one plate, and a left-handed thread for the other, so that when the spindle is revolved the plates move in opposite directions. The nuts should, of course, be made of bronze, and provided with collars or other suitable device, so that when the spindle is turned round they do not revolve with it, but only move the plates. The valve spindle usually passes through both ends of the valve box, and is connected to the eccentric-rod by a swivel-joint, which permits it to be turned round; the other end has a feather-way cut in it, and passes through a socket, which is held in place by a suitable bracket, provided with a fixed feather which fits into the groove in the spindle, and a wheel by which it can be turned; by these means the spindle can be turned while the engine is at work.

This plan suffices for small engines, but the power required to move the spindle rapidly increases with the size of the engine, and this not so much owing to the friction of the valves or external fittings, as to that of the screwed parts of the spindle in the nuts on the back of the valve from want of lubrication. This difficulty is so great that in engines of 50 N.H.P., only after a few days of work without altering the position of expansion valve, it has been found impossible to turn the spindle round with the means at command. To avoid this difficulty the gear for varying the cut-off should be outside the valve-box, and under the inspection of the engineer. A good plan for carrying out this is to provide each valve plate with a separate rod; each rod passes through a stuffing-box, and is connected to a crosshead by a nut in it, which is free to turn round so as to adjust the position of the spindle. These nuts may have wheels keyed to them gearing into one another, or they may have worm-wheels operated on by a worm between them; in either case the nuts will turn round in opposite directions, so that if the threads on both the spindles are right-handed, the turning of the nuts will cause the valves to move in opposite directions. The spindles are secured to the valve plates, so that they cannot turn round when the nuts are turned; they are also of necessity out of centre with the valves, which must consequently be guided sideways on the main valve. The only objectionable feature in this arrangement is, that the spindles are not secured to the middle of the valves; but practice has shown that this is no detriment, and the whole system works exceedingly well.

If the engine to which this kind of expansion valve is fitted, is required to work with the same efficiency in *stern-gear* as in *head-gear*, the eccentric should be in line with and on the side of the shaft opposite the crank, and its throw should not be less than that of the main valve eccentrics. If, however, the engine will seldom, and for only short periods, work in *stern-gear*, the eccentric should be on the same side as the crank, but nearer the *stern-going* eccentric than the *head-going* one; then the relative travel of the valves is greater in *head-* than *stern-gear*, and the cut-off is also earlier in head-gear, so that if a sudden order were given to *astern*, the engine is better prepared for carrying out that order.

The equalisation of cut-off at top and bottom of cylinder, for all positions of cut-off, is effected by causing one plate to move more per revolution of the spindle than the other; and in the case of the double spindle arrangement, the wheels on the nuts are so proportioned as to obtain the same result.

Inside Cut-off Valves.—So called, because steam is cut off at the inside edges of the expansion valve. The eccentric is in this case, also, nearly in line and on the same side as the crank, so that the relative travel with the same throw of the eccentrics is greater than in the case when the expansion eccentric is opposite the crank, and consequently on the same side as the eccentrics of the main valve.

The variation in cut-off can, in this case, be effected in a manner similar to the last; but since the relative travel is so much greater, and since it would be inconvenient to spread out the valve so much, it is usually effected by varying the travel by means of a link, as already described, or by a link having one end connected to an eccentric and the other to a rod jointed to a fixed point on the foundation, or more usually on the shaft, and called the *dumb* rod, because of its similarity in position to the eccentric-rod, without the longitudinal motion of the latter.

Piston Expansion Valves.—When the main valve is a piston valve, the expansion valve may be also a piston valve working within the main valve, and on the same principles as described for slide valves. This plan has been put into practice by several engineers, and although somewhat complex in structure, does, no doubt, work satisfactorily, and with little friction. The spindles are sometimes out of centre; but not always, as the expansion valve-rod passes through the top of the valve-box, while the main valve-rod is through the bottom as usual.

Expansion Valves for Compound Engines.—The Admiralty, in order to carry out the principle, that to work a compound engine with maximum economy there should be as little *drop* as possible, and that the power shall be evenly divided between the two cylinders, used to fit expansion valves to both cylinders of even very small engines, and thereby so complicated them as to render them anything but easy to keep in order and to work. Some English engineers, and very many French engineers, always provided an expansion valve to the high-pressure cylinder, so that when working at high grades of expansion, nearly the whole of the work was done in that cylinder. It has been stated that to effect a cut-off earlier than half-stroke *efficiently*, a separate cut-off valve is necessary, but this is only strictly true when the full speed of the engine is maintained with such a cut-off. If full speed is attained at a cut-off somewhat later than half-stroke, a *reduced* speed can be obtained *with efficiency* with an earlier cut-off by simply "linking up;" for the port area being suited to the higher speed, the reduction of opening on "linking up" is not materially felt at the reduced speed, the consequent compression also adds to the economic working of the engine at high grades of expansion, and if the valves have been so set that at full speed the work is evenly divided between the cylinders, so it will be, with but slight variation, on "notching up" the links of all the cylinders (*v.* fig. 106). Expansion valves are needless additions to the marine engine, and should be avoided, as all sources of possible breakdown or derangement ought. These valves have never been fitted to triple-expansion engines.

Eccentrics.—The sheaves, or, as they are sometimes called, pulleys, are made usually of cast iron, and, when possible, each is in one piece, bored out so as to fit the shaft on to which it is keyed. It is essential that it shall fit the shaft quite tightly, or otherwise it will soon become loose from the continual sudden and intermittent application of the strain. When the couplings or flanges on the shaft do not admit of the sheaves being fitted in this way, it is usual to divide them on a line through the centre of the shaft at right angles to the line passing through the centre of shaft and centre of eccentric. The two unequal parts are securely bolted together, and keyed to the shaft on the line through the centres, so that the whole of the strain comes on the larger and stronger half, the lesser half acting only the part of a clamp to hold it to the shaft. When it is desirable to keep the diameter of the sheaves as small as possible, this clamp piece should be of wrought iron or steel. Great care should be exercised in fitting the two parts of the sheave together, and also in "bedding" them on the shaft. Some engineers make all eccentric sheaves in parts, owing to the difficulty of fitting them to the shaft when in one casting. Sheaves are sometimes divided through the line of centres, so that each part is a half, but this is not so satisfactory a plan, as, from want of a connection at the small part, the joint is very apt to spring open and damage the strap, besides eventually causing the sheave to become loose on the shaft. When such a division is made there should

S.S. "KOVNO" CYLINDERS, 25 AND 50 × 45.

	Steam.	Vacuum.	Revs.	Mean Pressure.		Indicated Horse Power.		
				H.P.	L.P.	H.P.	L.P.	Total.
1st Grade,	87	25½	73	59·92	11·62	430	379	809
2nd Grade,	90	26½	64	35·8	9·07	256	259	515
3rd Grade,	88	27	39	18·0	4·57	78·3	79 5	157·8

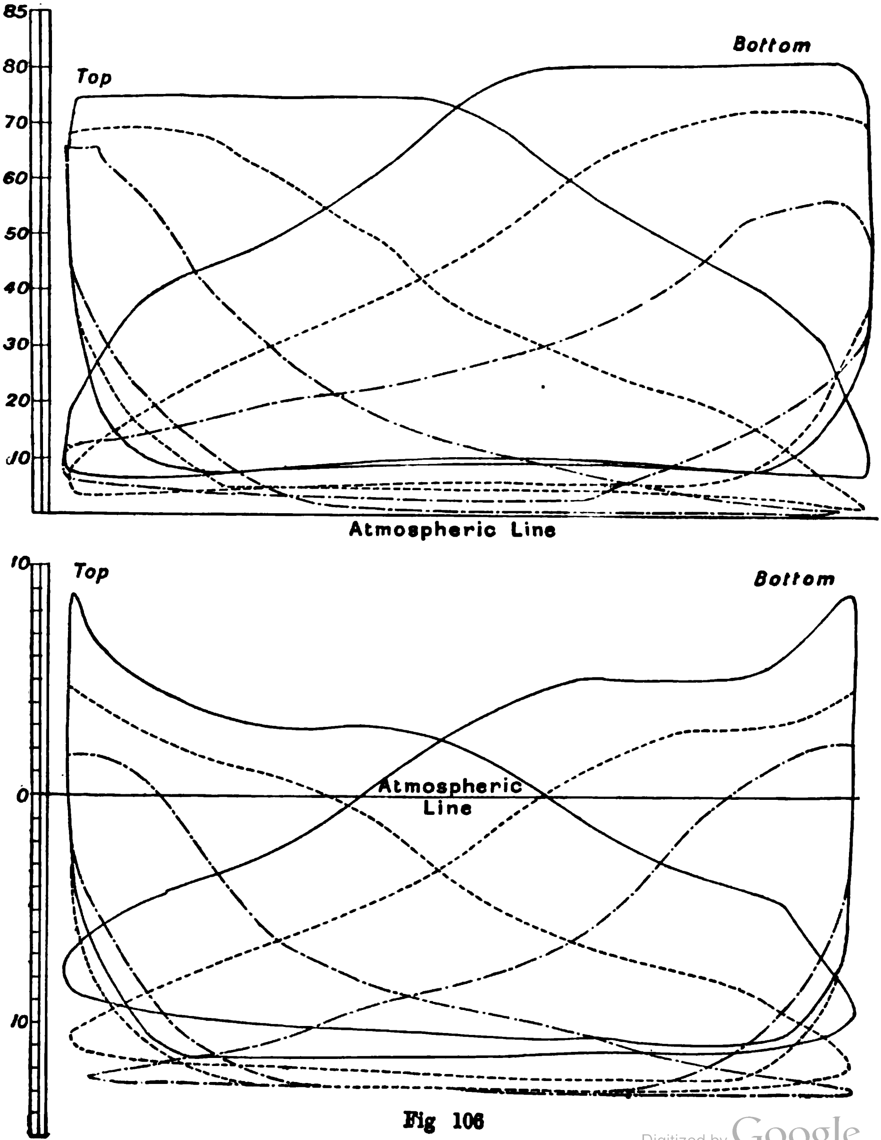


Fig 106

be a connecting bolt as close to the shaft as possible, as well as one at the extremity, and also there should be a key to each half of the sheave and the shaft. When the eccentrics are very large, and have to drive heavy valves, it is also advisable to make the small part, when divided, as above described, of wrought iron or cast steel.

The diameter of an eccentric sheave = 1·2 (throw of eccentric + diameter of shaft).

Breadth of the sheave at the shaft - - = 1·25 × D + 0·65 inch
" " " strap - - = 1·2 × D + 0·63 "
Thickness of metal around the shaft - = 0·7 × D + 0·5 "
" " at circumference - = 0·6 × D + 0·4 "
Breadth of key - - - - - - = 0·7 × D + 0·5 "
Thickness " - - - - - - = 0·25 × D + 0·5 "
Diameter of bolts connecting parts of strap = 0·6 × D + 0·1 "

The headgoing sheave should be 33 per cent. broader.

D is found as before, and is = $\sqrt{\frac{L \times B \times p}{F}}$. (*v.* p. 278).

Eccentric Straps are made generally in the form of a hoop, with lugs to connect the halves together, and with a base for the rod, if it is not formed with the strap. When the face of the sheave is formed with small flanges on each side, the section of the strap is rectangular; but the more common practice is to make the strap with small flanges, so as to overlap the face of the sheave, and bear on parts on each side. The advantage of the latter plan is that the sheave is not necessarily broader than the strap, and the oil does not run off so readily when the engine is at work.

Some engineers prefer to turn the face of the sheave with a V-shaped groove, and form the strap to fit into this; but it is not a good plan, owing to a small amount of wear permitting of very considerable side play (this again, though generally a fault, becomes a virtue with badly-fitted link-motions).

Eccentric sheaves have been made with a narrow projecting collar in the middle of the face, the straps having a groove turned to suit it. By this plan the straps are prevented from having side play, but the oil is not retained, as is the case with all the other methods.

The straps are usually made of bronze, or of wrought iron lined with bronze or white metal; but cast-iron straps give very great satisfaction, especially when of large size; cast iron lined with white metal also has been adopted by some engineers with success. Malleable cast iron and cast steel, lined with either brass or white metal, while having a strength beyond that of the cast iron, do not cost nearly so much as bronze or wrought iron for straps, and are now being largely employed in the mercantile marine.

When of bronze or malleable cast iron :—

The thickness of eccentric strap at the middle = 0·4 × D + 0·6 inch.
" " " sides = 0·3 × D + 0·5 inch.

When of wrought iron or cast steel :—

Thickness of eccentric strap at the middle = 0·4 × D + 0·5 inch.
" " " sides = 0·27 × D + 0·4 inch.

Eccentric-rods.—Unless these are very short indeed, it is better to make them separate from the straps, as then the breakage of either does not condemn both. They are made of wrought iron when long; steel cannot here be employed with advantage, as owing to its modulus of elasticity being so near that of iron, no reduction in size can be made.

The diameter of the rod in the body and lower end may be calculated in the same way as that of a connecting-rod, the length being taken from centre of strap to centre of pin.

The diameter of eccentric-rod at the link end = 0·8 D + 0·2 inch.

Eccentric-rods are, however, often made of rectangular section, which is the correct form for the stresses it has to withstand, but is not so economic to manufacture.

Reversing Gear should be so designed as to have more than sufficient strength to withstand the strain *of both the valves and their gear at the same time* under the most unfavourable circumstances; it will then have the *stiffness* requisite for good working.

Assuming the work done in reversing the link-motion, W, to be only that due to overcoming the friction of the valves themselves through their whole travel, then if T be the travel of valves in inches; for a compound engine

$$W = \frac{T}{12}\left(\frac{L \times B \times p}{5}\right) + \frac{T}{12}\left(\frac{L^1 \times B^1 \times p^1}{5}\right), \text{\&c.}$$

and for an expansive engine with two cylinders

$$W = 2 \times \frac{T}{12}\left(\frac{L \times B \times p}{5}\right); \text{ or } \frac{T}{30}\left(L \times B \times p\right).$$

To provide for the friction of link-motion, eccentrics, and other gear, and for abnormal conditions of the same, take the work at one and a-half times the above amount.

To find the stress at any part of the gear having motion when reversing, divide the work so found by the space moved through by that part in feet, the quotient is the stress in pounds; and the size may be found from the ordinary rules of construction for any of the parts of the gear.

CHAPTER XIV.

VALVE DIAGRAMS.

Motion of the Piston.—Fig. 107 illustrates an ingenious method proposed by Professor Zeuner for finding the position of the piston at any position of the crank, and should be used always when constructing a valve diagram, in order to find the points of cut-off, release, compression, &c.

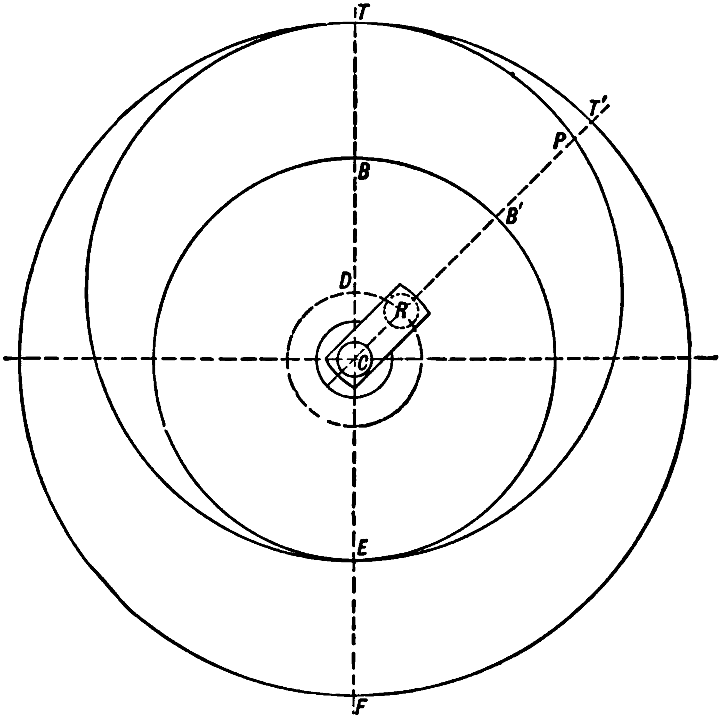


Fig. 107.—Diagram of the Piston Path.

The following is the construction of such a diagram :—Draw a line T F, and on it take a middle point C ; cut off a part C D, equal to the radius of the crank (to any convenient scale), and a part D T equal to the length of the connecting-rod.

With D as centre, and D T as radius, draw a circle cutting the initial line at E. With C as centre, and C T as radius, draw another circle ; and with C as centre, and C E as radius, draw a third circle. To find the position of the piston, with respect to its extreme position, for a position of crank C R when it has moved through an angle D C R from the "dead" point C D; produce C R to cut the circles at B′ P T′; then the piston has moved through a space T′ P in turning the crank through the angle D C R, and it is distant from the other end of its stroke by a space P B′. The correctness of this construction is easily seen by supposing an engine whose piston is connected directly to the crank-pin (such as the trunk engine) to be turned around about its shaft, while the crank remains stationary in the position C D. The circle T T′ F represents the path of the back end of the cylinder, and B B′ E that of the front end, since the cylinder is turned about the point C; the path of the piston will be on the circle T P E, since it is held at the same distance from the crank-pin D.

Diagram for the Common Slide-valves.—(1) Given the travel of the valve, amount of the lead or opening of valve at commencement of stroke, and the point of cut-off, to find the lap of the valve and the position of the eccentric, &c. These are the conditions generally predetermined in every-day practice.

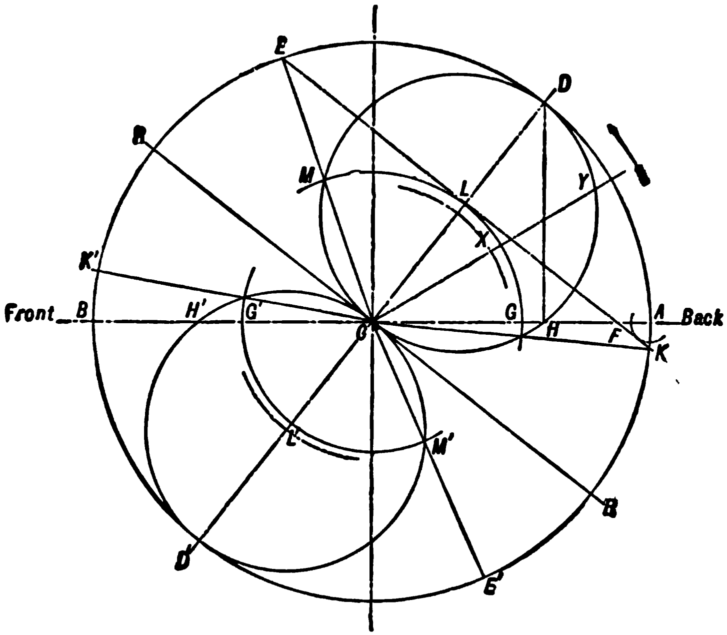


Fig. 108.—Zeuner's Diagram for the Common Valve Motion.

Fig. 108.—Draw a straight line, whose length A B is equal to the travel of the valve, and on it as diameter draw a circle, the centre of which is at C. Draw a line C E, so that A C E is the angle through which the crank has to move to arrive at the position of cut-off (this is to be obtained by drawing the piston diagram outside the circle A E B). With A as centre, and a radius A F equal to the *lead*, draw part of a circle, and through E draw a line touching the circle and cutting the original circle at K.

Through C draw a line C D perpendicular to E K, and cutting it at L. This line C D bisects the angle K C E.

C L is the amount of lap required, and B C D is the angle between the crank and eccentric to obtain the cut-off required; and since C D is the half-travel of the valve, L D is the maximum amount of opening of the port.

To extend the usefulness of the diagram, on C D as diameter, draw a circle, and with C as centre, and L as distance, draw the arc of a circle G L M, which is called the lap circle. The part G H intercepted by these circles, is equal to A F, and represents the *lead* or opening when the crank is in position C A; X Y likewise represents the opening of the port, when the crank is in position C Y.

To determine the operation of the valve at the other end, it is necessary to produce D C to D′ and on C D′ as diameter, draw a circle. Let H′G′ be the "lead" at the other end of the valve; with C as centre, and C G′ as radius, draw another lap circle, cutting the circle C D′ at M′. Through C M′ draw a line C E′. Then C E′ is the position of crank at cut-off, it having moved through the angle B C E′ and C G′ is the amount of lap required.

(2) If, however, the *travel*, *lap*, and *lead* are given to find the position of eccentric and cut-off, the following is the construction:—

Draw on A B as diameter, the circle as before. Cut off C G equal to the lap, and G H equal to the lead. At H, erect a perpendicular to A B, cutting the circle at D. Join C D, and on it as diameter, draw a circle; with C as centre, and C G as radius, draw the lap circle, cutting the circle C D at M. Draw a line through C M, cutting circle A D B at E.

Then C E is the position of cut-off, and B C D is the angle between the crank and eccentric.

(3) Given the *travel*, *lap*, and *position of eccentric*, to find the cut-off, lead, &c. Let B C D be the angle between the crank and the eccentric. Draw the travel circle as before, and on C D as diameter draw the circle cutting A B at H. Draw the lap circle G L M, then C M E is the position of cut-off, and G H is the lead.

The position of the crank when the valve commences to open is C K, or at the angle A C K, with its initial position.

If the valve has no inside lap—that is, the ports are both just closed to exhaust when the valve is in mid position—the position of release is at C R, a line at right angles to C D; and the position at which compression takes place is C R′, also at right angles to C D.

If, however, the valve has *inside* lap amounting to, say, C G′, release will not take place till the crank is at C K′.

If, on the other hand, the valve is cut away on the inside so as to have negative lap to the amount of C M, then release will take place at C E, and compression at C K.

If the inside lap is less than these amounts, the position of release and compression can be found by drawing an inside lap circle with a radius equal to the lap, and through the points where it cuts the circle on C D and C D′, the lines drawn through C and those points of intersection will give the positions of crank.

Travel of Valve.—It is usual to fix the travel of the valve before determining any further particulars, as so many things depending on this have often to be considered before there is leisure to finally decide the lead, lap, &c.

It will be seen, on reference to the diagram, that the opening to steam will vary with the travel, and if the *area* of opening is fixed from certain considerations before-mentioned, it is an easy matter to calculate how much the valve must open to give that area. Now the opening, together with the lap, is equal to half the travel, and with the same *positions* of "lead" and cut-off, the opening will be a constant ratio of the travel. This ratio can be determined by drawing a preliminary diagram with an assumed travel of valve. Let R be this ratio, and Q the amount of opening desired; then

$$\text{Travel of valve} = Q \div R.$$

Since it is usual to design the ports of a steam cylinder so that the flow of steam when exhausting may not exceed a certain velocity, it is evident that the port should open fully for that purpose; hence the travel of the valve, when there is no inside lap, should not be less than twice the length of the port, and is generally about 2½ to 3 times the length of the ports.

The less travel the valve makes, the less work is absorbed in moving it, as the work is very nearly proportional to the travel. Double- and treble-ported valves are resorted to with the object of reducing the travel, as by them double and treble openings are obtained, and the travel may, therefore, be one-half and one-third that of the common valve with single ports.

"Lead."—The amount of lead given to the valve is generally decided arbitrarily and according to judgment or prejudice. It will be seen by referring to fig. 108 that with the same "lap" an earlier cut-off is obtained by moving the eccentric farther from the crank, but at the same time increasing the "lead;" if the earlier cut-off is to be obtained without altering the "lead," the lap must be increased and the eccentric moved; but if the lap be increased, the opening is decreased, and the resistance at the port thereby increased. If, therefore, an early cut-off is required without the aid of an expansion valve, the "lead" must be great or the travel great, to get sufficient opening to steam, or the lead and travel must be both larger than would be the case with a later cut-off.

The considerations which should operate in deciding the amount of lead are the speed of the engine and the weight of moving parts compared with the piston area. The momentum of the piston and rods should, if possible, be absorbed in compressing the steam remaining in the cylinder when the valve closes to exhaust; and were this always the case, there would be no need for fresh steam to enter the cylinder until the piston was at the end of its stroke; but this, under ordinary circumstances, seldom occurs, and if no fresh steam were admitted, "to form a cushion" for the piston, there would come a considerable jar on the bearings, &c., at the end of every stroke, owing to the strain increasing the displacement of the shaft in its bearings, and the sudden application of the load when the valve opens, causing its replacement with considerable force. This action is also observable when there is considerable "lead" without adequate "compression;" and although the "hammering" is usually attributed to excessive "lead," it is really due to want of compression, for on "notching-up" the link, it generally ceases, notwithstanding that the lead is then thereby very considerably increased.

Long-stroke engines may have considerably more lead than those of the same cylinder capacity with shorter stroke, as the weight of moving parts is not less and sometimes more, the piston velocity is more (with the same number of revolutions), while *the piston area is less*. Fast-running engines may also have more "lead" than slow-running ones, as the element of time bears on the operation of the steam. Again, if owing to the valve motion the valve opens slowly, the period of admission may be earlier than should be the case with a quick opening one, as the wire-drawing at the commencement of admission will produce a similar effect to compression.

The amount of lead at each end will vary for two reasons; first, the cut-off at each end being effected by the same eccentric, any variation in position of cut-off must be obtained by varying the lap, and any variation in the lap will cause an inverse variation in the lead; secondly, the lead should be less at that end of the valve remote from the gear, as the adjustment after wear of the latter tends to increase the "lead" at that end. If the cut-off is equal at both ends the laps will vary, that at the end of a direct acting engine remote from the shaft being more than that at the other. With the ports as generally found in every-day practice, and with such travels as are practicable, the cut-off at the end near the shaft should be earlier than at the end remote from it, as the larger opening got by the decreased lap causes a fuller indicator-diagram at that end. For these reasons many engineers arrange so that the lead at the back or top end is only half that at the front or bottom end.

When a compound engine is to be worked with a cut-off at or before half-stroke without expansion valves, it is best to arrange the valves to cut off at about 0·6 of the stroke with very small "leads," so that when required the engine is easily handled, and when working at its normal speed the earlier cut-off is obtained by notching up without excessive leads.

Inside Lap.—The effect of positive lap on the exhaust side of the valve is to retard the period of release, and to increase the compression by accelerating the closing to exhaust. The effect of negative lap is, of course, the reverse of this, and is also to give a full opening to exhaust sooner than would be the case with positive lap. A large amount of lap, either negative or positive, is necessary on the inside of a valve to materially change the periods of release and closing to exhaust, because just then the valve is moving at its quickest speed; but a small amount of negative lap makes a considerable difference to the amount of opening to exhaust at the end of the stroke, when the steam should exhaust wholly from the cylinder. There is no advantage in retaining steam in the cylinder to the very end of the stroke, as the effort of the piston on the crank is ineffective to produce good results, and any forward pressure only serves to increase the difficulty of bringing the piston gently to rest by means of cushioning, &c., and this is especially true of quick-moving engines, whose steam ports are comparatively small.

If the valve has negative lap on the exhaust side, it is manifest that at a certain period there is a communication between both ends of the cylinder; when the negative lap is considerable, and the ports small, the effect of such a communication is to fill the exhausted cylinder on one side of the piston with the steam released from the other side, just before the valve closes to exhaust, and is seen very distinctly on the indicator-diagrams, in the form of a sudden rise of pressure at the commencement of the compression. In cases of this kind, ample compensation is made for the retarding of the compression by negative lap in the increase of back pressure at the commencement of compression.

The high- and medium-pressure valves of compound screw engines should always have a considerable amount of negative inside lap, to allow of a continuous and easy flow of steam during exhaust. The amount of negative lap on the inside should be *always* less than the outside lap of the valve, otherwise there will not be enough bar to cover the port, so that steam can pass from the valve-box to the exhaust just as the valve is opening and shutting; for a quick-running compound engine there should be such lap that exhaust commences from the high- and medium-pressure cylinder at 0·9 of the stroke; and with a slow-working paddle engine not later than 0·95 the stroke.

Effect of "Notching Up."—When an engine is fitted with link-motion or other means of reversing it without throwing "out of gear," so that the mechanism which would produce stern motion, can be made to effect that producing ahead motion, a certain variation in the cut-off, release, &c., can be effected by what is called "notching up." The origin of this term is clearly traceable to the locomotive, whose reversing lever is held in place by a sliding-bolt fitting into a *notch* in a quadrant provided for the purpose.

When the link is notched up so that the block works the valve from a point between its two extreme positions on the link, the motion is one due to the combined effort of both eccentrics, and in order to examine clearly the operation of the valve under these circumstances, it is necessary to find the position and throw of the *equivalent eccentric.* To determine this exactly is somewhat difficult, but a very close approximation can be made by the method suggested by Mr. Macfarlane Gray, as follows:—

Suppose the link (fig. 110) to be notched up to a point M; or, in other words, the link moved so that the link block is distant M T from the point at which the eccentric-rod is attached to the link.

Draw the valve diagram (fig. 109) due to the position and throw of the eccentrics of fig. 110; and through D D′ draw the arc of a circle with a radius found as follows:—

Radius = length of eccentric-rod from centre to centre × half the distance between the centres of the two eccentric sheaves ÷ the distance between the centres of eccentric-rod pins on the link.

$$\text{Referred to fig. 110,}\quad \text{Radius} = \frac{\text{D T} \times \text{D D}'}{2\,\text{T N}}.$$

$$\text{On this arc, take a point Z, so that}\ \frac{\text{D Z}}{\text{D D}'} = \frac{\text{T M}}{\text{T N}}.$$

Join C Z, and on it as diameter draw a circle, cutting the lap circle at points L and E; through C L and C E draw lines which represent the position of opening and closing respectively at one end of the valve when the link is notched up to the point M.

It will be seen that the effect of notching up is the same as would be produced by an eccentric whose position on the shaft is at the angle B C Z with the crank, and its eccentricity C Z; that the opening to lead is earlier, and the magnitude of the lead considerably increased; that the reduced travel gives a smaller opening to steam, and that the cut-off is earlier; likewise, by examining further, it can be seen that release takes place sooner and compression commences earlier than when in full gear.

When the eccentric-rods are arranged as shown in fig. 110, they are said to be *open*, or it is an *open rod link-motion;* if, on the other hand, D N and

D′T be joined, it is said to be a *crossed rod link-motion*. When the link motion has crossed rods, the arc D Z D′ should be convex to the point O, that is, the centre of curvature is on the side A.

When the rods are crossed, an earlier cut-off is effected by notching up without materially altering the *lead*, and, in most cases, with a very early cut-off there is only a very small *lead*, smaller even than when in full gear. But the travel is reduced very rapidly by notching up, giving a corresponding reduction of opening to both steam and exhaust; so that, altogether, crossed rods are not so convenient as open ones for working expansively by notching up the link, and are very seldom so fitted.

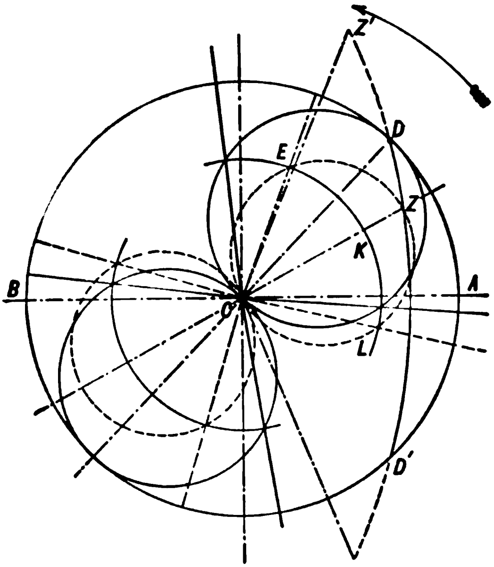


Fig. 109.—Diagram showing the Effect of "Notching up."

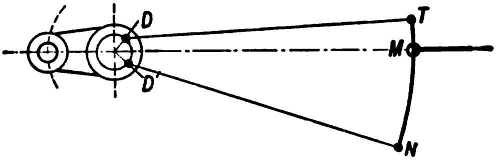


Fig. 110.—Link Motion "Notched up."

The valve diagrams, when the link-motion is such as shown in fig. 110, are constructed on the same principle as for notching up, and the position of the eccentrics and their throw are determined by producing the arc D D′ (fig. 109), and taking a point Z′ *beyond* D, so that

$$\frac{Z'D}{DD' + 2\,Z'D} = \frac{TM}{TN}.$$

Join O Z′.

Then C Z′ is the eccentricity of the sheaves, and B C Z′ is the angle between them and the crank.

Expansion Valve on an Independent Face, Central Position Ports Closed.—The valve in this case is working under precisely the same conditions as a common slide-valve, and the same construction of diagram as fig. 108 holds good.

Expansion Valve on an Independent Face, Central Position Ports Open.—Let C O′ (fig. 111) be the position at opening of valve when set at earliest cut-off, which should be somewhat before that of the main valve, and C E the position at the earliest cut-off which is required. Bisect the angle O′C E by the line C T, and make C T equal to half the maximum travel of the valve.

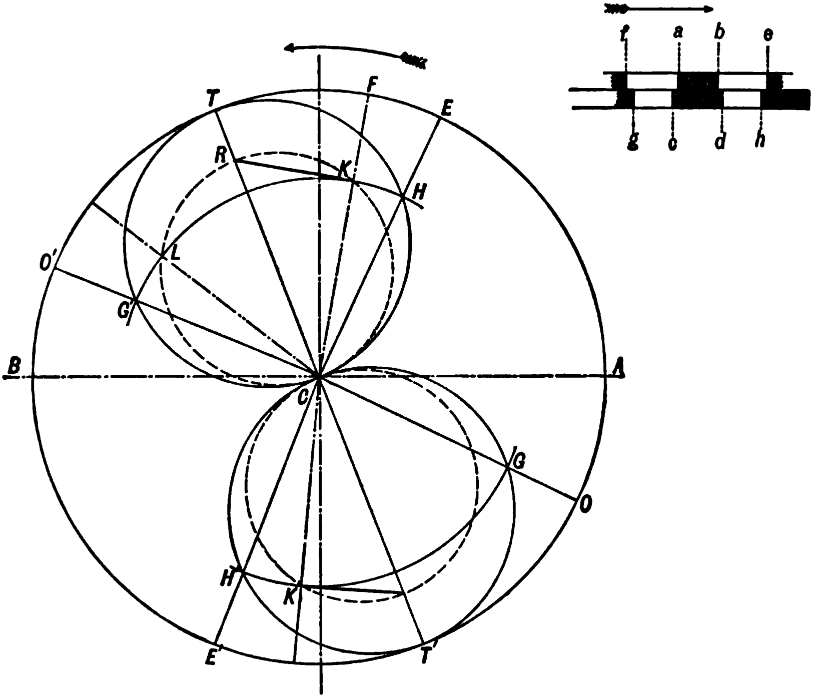


Fig. 111.—Diagram for an Independent Expansion Valve.

On C T as diameter, draw a circle cutting C E at H.

Produce C T to T′, making C T′ = C T.

On C T′ as diameter, draw a circle cutting C O at G.

Then C H is the distance of the cutting-off edge of the valve from the edge of the steam port; that is, C H = *b h*; and the lap of valve *b d* = C H − length of port.

The width of the bar *a b* must not be less than C T − *b d*, and should be = (C T − *b d*) + ¼ inch.

For the other end of the valve, a similar construction will give the required results.

Let O E′ be the position of cut-off at the other end of the cylinder, cutting the circle O T′ at H′.

With O as centre, and O H′ radius, draw a circle cutting the circle C T′ at G; through C G draw a line, then C G is the position at which the valve opens again.

Hence *a g* = C H′; and the lap *a c* = C H′ − length of port.

Again, the width of bar *a b* must not be less also than C T − *a c;* and should be = (C T − *a c*) + ⅛ inch.

The bar *c d* = *a b* + lap *b d* + lap *a c.*

To find the reduction in travel for a later cut-off at position O F cutting the port circle at K. Draw K R perpendicular to O F cutting C T at R. On C R as diameter, draw a circle cutting the lap circle at L. C R is the half travel of valve. O L is now the position at which the valve opens again, which should not be before the main valve is closed to steam.

In arranging an eccentric and gear for this valve, care must be taken in setting it that the expansion valve will open at the earliest cut-off before the main valve opens, and that at the latest cut-off it does not open again till the main valve is closed. When the expansion valve spindle is in the same plane with the piston-rod, and the gear is direct from the eccentric to the valve-rod, the eccentric should be set at the angle B O T with the crank on the same side as the *head-going* eccentric.

Expansion Valve Working on the Back of the Main Valve, Cutting off at the inside edge, and Variation in Cut-off made by Varying the Travel.—Let A (fig. 112) be the dead point of the crank at the back of the stroke, and C E the position of earliest cut-off. B C D is the angle between the crank and *head-going* eccentric, and C D its eccentricity. Cut off C H equal to the lap of the expansion valve, or distance of the inner edge of expansion valve from inner edge of steam port in the back of the main valve, when the expansion valve is in its mean position with respect to the main valve. Draw the lap circle H G as before. From H draw a line H O perpendicular to C E with D as centre and the half travel (absolute) of the expansion valve as radius, draw the arc of a circle cutting H O at O. Join D O and C O; on C O as diameter, draw a circle, which will pass through H, since C H O is a right angle. Complete the parallelogram C D by drawing D T parallel to O C, and C T parallel to D O. Then C O is the half-travel of the expansion valve relative to the main valve; B C T is the angle between the crank and the expansion eccentric, or T C D is the angle between the expansion eccentric and the head-going eccentric.

By producing O C to O′, and on C O′ drawing a circle, &c., the cut-off, lap, &c., may be investigated for the other end of the valve.

To find the effect of shortening the absolute half-travel to C R. Since the position of the eccentric remains unchanged, join R D, and through C draw C N parallel to R D, and cutting O D in N. On C N as diameter draw a circle cutting the lap circle at L and K.

Through C L and C K draw lines which are respectively the positions of opening and cut-off of the expansion valve for the back end, and C N is the relative half-travel.

It will be seen that when the travel of the expansion valve is nothing, that the relative half-travel is C D, and that if the lap is the same as that of the main valve, the cut-off and lead will be the same also. For this reason it is customary when designing a valve of this kind, which may have to go out of gear when at full speed, or for some particular purpose, to make the lap the same as that of the main valve, and for practical reasons to make the travel the same also.

To find the effect of the expansion valve when in stern-gear, join T to D′ of the stern-going eccentric, and complete the parallelogram C D′ as before.

It will be seen also, that with this kind of expansion valve, the opening to steam is never less at the expansion valve ports than at the cylinder ports.

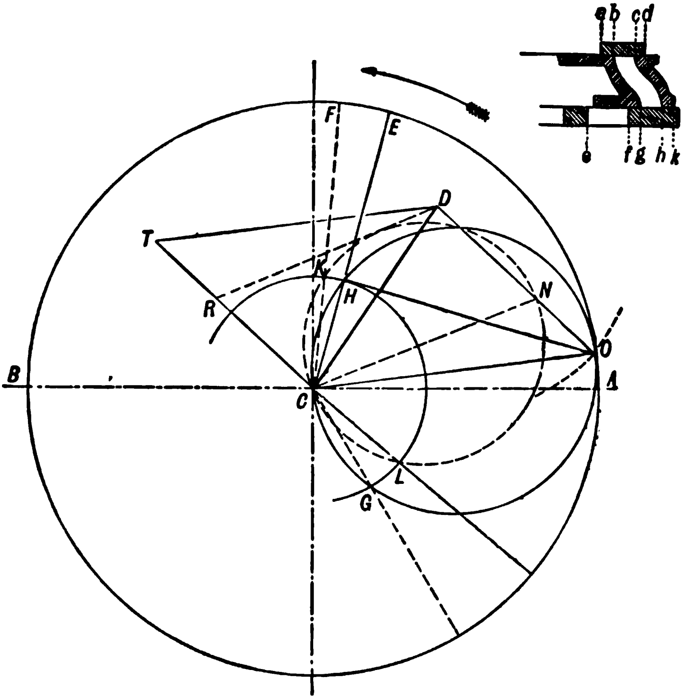


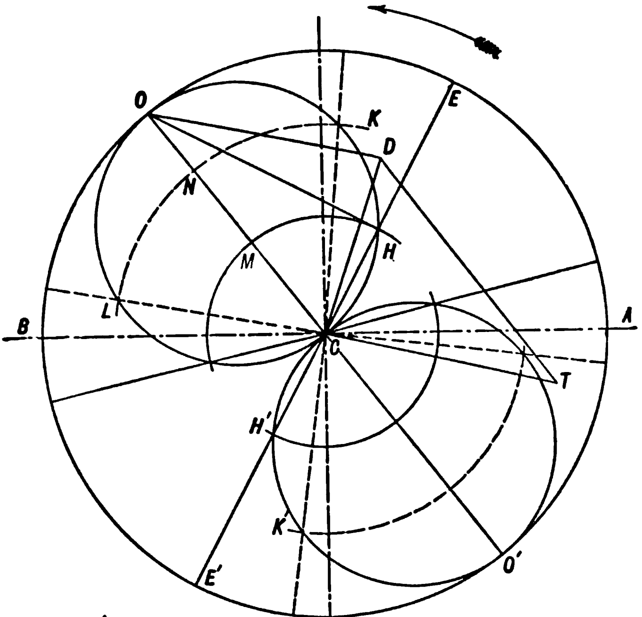
Fig. 112.—Diagram for an Expansion Valve.

Expansion Valve Working on Back of Main Valve, and Cutting off at Outside Edge.—Let A (fig. 113) be the dead point of the crank, as before, at back of stroke, and C E the position of earliest cut-off required, B C D is the angle between the crank and the head-going eccentric, and C D its eccentricity. From C E cut off a part, C H, equal to the opening to steam of the main valve at that end, or such as would give sufficient opening by the rules as laid down already. From H draw a line, H O, perpendicular to C E. With D as centre, and D O a radius equal to the eccentricity of the expansion eccentric, draw the arc of a circle cutting H O at O. Join C O, and on it as diameter draw a circle, which will pass through H, since C H O is a right angle.

Complete the parallelogram C D by drawing D T parallel to C O, and C T parallel to O D.

Then C O is the half-travel of the expansion valve with respect to the main valve; B C T is the angle between the crank and the expansion eccentric, and T C D the angle between the expansion eccentric and the head-going one of the main valve. C H is the distance of the edge of the expansion valve (fig. 114) from the outer or cutting-off edge of the port on the back of the main valve, when the expansion valve is in its mean position with respect to the main valve.

If the expansion eccentric is set *exactly* opposite to the crank, so that the point T is on the line A B, then the cut-off is the same in *stern-going* as in *head-going* gear. If, however, it is set as shown in fig. 113 so as to be nearer the stern-going eccentric, the cut-off is later in *stern-going* than in *head-going gear*, and the relative travel of the expansion valve is greater.



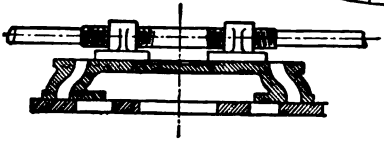


Fig. 114.—Expansion Valve. Fig. 113.—Diagram for an Expansion Valve.

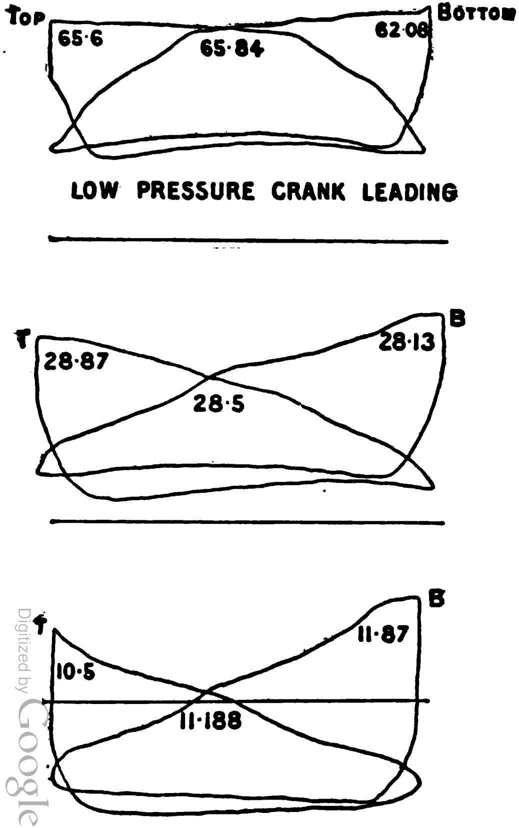
If it be required to cut-off at a later period, as at C K, the expansion valve must be so moved that when in mean position with respect to the main valve, the cutting-off edges are apart by a distance equal to C K; that is, the valve is moved towards the middle by a distance M N or C K − C H.

The laps, cut-off, &c., at the other end of the valve may be investigated in a similar way by producing O C to O′, making C O′ equal to C O, and on it as diameter drawing a circle cutting the required position of earliest cut-off C E′ at H′.

S.S. "Flamborough."

Cylinders 22, 35, 58 ins. diameter.
Stroke 42 ins.

	Low P. Leading.	High P. Leading.
Boiler press., lbs.	150	150
Vacuum, . ins.	26	26
Intmdt. recvr. press., . lbs.	54	58
Low recvr. press., . lbs.	8	7
Revolutions, .	67	62
Mean press., H.P.	63·8	65·9
,, M.P.	28·5	32·8
,, L.P.	11·19	10·12
,, referred,	30·73	31·55
I.H.P., . H.P.	344·8	329·4
,, . M.P.	389·6	415·1
,, . L.P.	420·1	351·8
,, . Total,	1154·5	1096·3



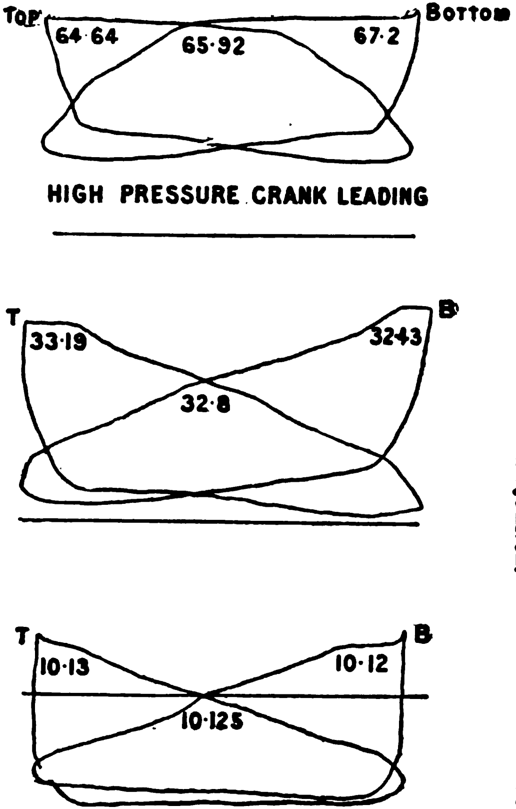


Fig. 115.—Effect of Crank Sequence.

C H′ is then the distance apart of the cutting-off edges of the valves at the other end. Also, if a later cut-off C K′ is required when C K is the cut-off at the other end, the valve must be moved towards the middle by a space equal to C K′ - C H′.

If C K be the position of latest cut-off, care must be taken that the arrangement is so designed that the re-opening of the port at C L is *after* the main valve has cut off to steam.

In order to obtain as large a relative travel as possible to the expansion valve, it is customary to make the angle B C D about 100°, so that the main valve has of necessity very little lap and small lead, and consequently there is a late release and a very small amount of compression.

In speaking of C D as the eccentricity, and B C D the angle of the head-going eccentric, it must be understood that an eccentric acting directly and in line with the main valve spindle is meant. If the valve gearing is such that the eccentric is otherwise, then C D is the eccentricity and B C D is the angle of the *equivalent eccentric.*

This remark applies likewise to the case of the expansion valve (fig. 114) cutting-off at the inside edge.

Construction of Valve Diagrams.—In making the diagram for an engine, the piston diagram (fig. 107) should be drawn first to such a scale as to be well clear of the valve diagrams. The main-valve diagram should then be drawn full size in black ink, and the expansion-valve diagram on it in red, or other distinctive colour. The effect of each valve can then be traced both separately and conjointly.

Effect on Indicator Diagrams of Crank Sequence.—Fig. 115 shows the effect of high-pressure crank leading the mean crank, and also the effect of the low-pressure crank leading the mean-pressure. It will be seen that in the former case the loads on the pistons are greater at the beginning of the stroke than is the case when the low-pressure leads. This is a good thing, especially in fast-running engines where the inertia of the moving parts is considerable; the ratio of maximum to mean load is greater in this case, and the variation in temperature is greater than when the low-pressure leads.

CHAPTER XV.

PROPELLERS.

THE fundamental principle on which all propellers act, is that the reaction caused by projecting a mass of water in one direction produces motion of the ship in the opposite direction.

If a ship is at rest with respect to the water in which it floats, and is to be set in motion by a stream of water projected from it, the inertia must be first overcome, and until this is accomplished the stream of water issues at the same velocity with respect to the water that it does to the ship, and its *slip* or motion with respect to the water is, therefore, equal to 100 per cent. of its motion with respect to the ship. If the stream issues at a constant velocity from the ship, its velocity with respect to the surrounding water will decrease as the motion of the ship increases. The speed of the ship will increase until the maximum speed due to the propelling effort is arrived at, when it will remain constant so long as the stream continues to issue at a constant rate. The whole of the effective propelling force is then employed in overcoming the resistance of the ship, and that resistance multiplied by the speed of the ship must be equal to the net effective power of the engine.

Let V be the velocity of the stream of water with respect to the ship, and v the velocity of the ship, then

$$\text{The velocity of the stream with respect to the water} = V - v.$$

This is called the *slip* of the stream, or the velocity with which it moves over the water on which it is projected.

$$\text{The percentage of slip is therefore } \left(\frac{V - v}{V}\right) 100.$$

The propelling effect of this stream of water will depend on the mass projected in a given time; and if it is issuing from an orifice, it will depend on the area of the orifice and the velocity of issue. If the area of the orifice is decreased, the velocity must be increased, and any increase of velocity of issue will cause an increase of slip; *per contra*, any increase of orifice will permit of a decrease of velocity, and consequently a decrease of slip.

If the water be taken from the surrounding water, *whose velocity with respect to the ship* is v, the velocity imparted by the machinery is then $(V - v)$, which is the *slip*.

Then if M be the mass of water, projected in a second, and $(V - v)$ its velocity in feet per second, and W its weight in pounds, and g its gravity $(= 32)$.

$$\text{The momentum} = M\,(V - v) = \frac{W}{g}\,(V - v).$$

Now, if A is the area in square feet of the issuing stream, and V is the velocity of issue, the weight of a cubic foot of sea-water being 64 lbs.,

$$\text{Weight of water projected in a second} = A \times V \times 64.$$

Therefore,

$$\text{Momentum of stream} = \frac{A \times V \times 64}{32} (V - v)$$
$$= 2 A \times V (V - v).$$

And this is the measure of the propelling force, and is equal to the resistance of the ship in pounds.

Now the stream of water may be projected from the ship by means of a common pump, or by a centrifugal pump or turbine, as on Ruthven's plan, as carried out in H.M.S. "Waterwitch," or by an instrument like the pulsometer, which was called by the inventor the *Hydro-motor*. In either case, there must be an inlet valve and pipe, and an outlet valve of some kind and pipe. If the stream is a large one, large pipes must be employed, or else the water must be projected at a high velocity. Since very large pipes are not always admissible, high velocity must be resorted to, and the efficiency is thereby very much impaired. To reduce the friction of the pipes, &c., as much as possible, they may be shortened, until, finally, the propelling pump is placed close to the water, and pipes avoided altogether. A step farther is taken, and the pump, in the form of a screw, is placed outside the ship, so that neither entry nor emission orifice shall check the flow of water. When this has been accomplished, it remains only for inventors to enclose it in a tube "to check the loss from the centrifugal action of the water," to place it farther into the dead wood, "to protect it from danger," &c., and, finally, to place it in the hold with inlet and outlet pipes, &c., as at first.

Jet Propulsion—Ruthven's Plan.—The propeller in this case consisted of a turbine wheel, working on a vertical axis near the middle of the ship, and enclosed in the usual case. Water was supplied to it from an orifice in the skin of the ship, which should be so placed, when possible, that the flow of water due to the onward motion of the ship is not checked in any way, but allowed to go direct, and have the additional velocity imparted to it by the centrifugal action of the wheel. But as such conditions are in practice almost impossible, the efficiency of this propeller is very much impaired by the check given to the water at the inlet orifice, and by the friction in the pipes and passages. The discharge from the turbine should be, if possible, quite direct in the opposite direction to the motion of the ship; but here again is a practical difficulty, for if the delivery pipe is through the stern, a separate pipe will be needed to pass through the bow to obtain the sternward motion. In H.M.S. "Waterwitch" there were two delivery pipes, one to each side, with directing valves and nozzles, so that the stream could be diverted ahead or astern on either side independently. This gave the ship a degree of handiness possessed by no other vessel afloat, but reduced the efficiency of the propeller far below that of both the screw and paddle.

There is one other very strong objection to this form of propeller beyond its inefficiency, and that is, the enormous size of the pipes and valves for a moderate velocity of flow; the objection becomes an insurmountable one when the power is such as to obtain even the moderate speed of 12 knots, for the holes in the skin of the ship are then of so large a size as to require special construction in their neighbourhood, and the pipes so enormously large as to occupy more space than can be well afforded. The weight of water, too, is so considerable as to take very much from the carrying powers of the ship.

There are certainly some attractive features in the jet-propeller for special circumstances, such as absence of vibration and "thud," freedom from accident to the propeller from exposure to shot or collision, and capability of handling the ship from on deck without stopping the engines; but from the causes indicated above it failed to induce further trials to be made by anyone, and it is quite certain that it cannot compete with the screw.

Hydro-motor.—In this plan there is no real engine or propeller; the water is simply ejected from two cylindrical chambers alternately by the direct pressure of the steam on its surface, as in Papin's engine. When the one chamber is emptied, steam is shut off from the boiler, and allowed to condense; the water flows quickly into the vacuous chamber, and, when it fills, its communication is opened to the boiler. A ship was fitted out on this plan, and proved fairly successful, for the condensation of steam on the walls of the wet chamber is very slight, owing to the short time of exposure, and that on the surface of the water equally small, for there is no convection to carry away the heat into the body of the water. The steam was also worked expansively, and the inventor claims for it quite as high an efficiency, judged by coal consumption, as that of a good compound screw engine. The walls of the chamber were lined with wood, so that there might be as little loss as possible by conduction.

However much the loss from such causes, as stated, may be set off by the gain due to absence of mechanism (and it is possible that the saving may even outbalance the loss), there still remains the same objection which militates so strongly against jet propulsion—viz., the large pipes, valves, and passages.

The plans for ejecting a stream of water by means of pistons, floats, &c., are very numerous, and since most of them have been re-invented many times over, the patents are legion. None of them have, however, been proved to be equal in efficiency to the commonest of screws, and many of them are most egregious mistakes from want of knowledge of first principles.

Common Paddle-wheel, or Wheel with Radial Floats.—This, the simplest and oldest form of paddle-wheel, consists essentially of a wrought-iron frame wheel, having two or more sets of arms fitted to a cast-iron "hub" or "boss," and connected by rims and cross-stays so as to have the necessary rigidity and interdependence of the several parts. Flat boards are secured in a radial direction to the arms by hook bolts, so that when the wheel is turned around, the floats, as these boards are called, drive a stream of water through the still water in the opposite direction to that which the ship takes. If the water were unyielding, the action of the wheel would be analogous to that of a pinion on a rack in the mangle motion, so that the motion onward of the ship per revolution would be 3·1416 × the diameter of the *pitch circle* of the floats or circle of centres of pressure. But as the water yields to the pressure of the float, the action is the reverse of that of an undershot radial water-wheel; the float drives a stream of water equal in area of section to its own area, and with a velocity $(V - v)$, as before.

Let D be the effective diameter of the wheel, A the area of a pair of floats (one on each wheel acting at once) in square feet, R the revolutions of the wheel per second, and v the speed of the ship in feet per second.

Weight of a cubic foot of sea-water = 64 lbs.

The velocity of the floats per second $V = \pi \times D \times R$.

The quantity of water operated on in a second $= A\,(\pi \times D \times R)$ cubic feet, or $64\,(A \times V)$ pounds.

The velocity imparted to the water is $(V - v)$.

Taking gravity as 32,

$$\text{Momentum of water} = \frac{64\,(A \times V)}{32} \times (V - v)$$

$$= 2\,(A \times V) \times (V - v),$$

which is, as before explained, equal to the resistance of the ship and the force on the floats, and consequently the *thrust* of the shaft on the bearings on the side of the ship.

But this is only strictly true when the float is just immersed and vertical, as there are other actions which seriously interfere with the efficiency of the radial paddle-wheel.

The floats of a radial paddle-wheel of necessity enter the water obliquely under all circumstances, and when the ship is deeply laden, so that the wheel is deeply immersed, the obliquity is very great. The disturbance consequent on this is such as to raise a stream of water nearly parallel to the underface of the float, which is finally projected past the emerging floats and astern of the wheel, and appears to come from them, and so giving rise to the generally received opinion, that the cascade of water usually observed in wake of a radial wheel is that carried up by the floats in leaving the water.

On this account it is impossible to estimate with any degree of accuracy the actual momentum of the race of water from a radial wheel. Paddle-wheels with radial floats are now employed only for service in barbarous countries, where simplicity and a minimum risk of derangement is a necessity.

$$\text{The effective horse-power} = \frac{2(A \times V) \times (V - v) \times 60v}{33{,}000} = \frac{AVv\,(V - v)}{275}.$$

$$\text{I.H.P.} = \frac{AVv\,(V - v)}{E \times 275}.$$

If E (the efficiency) be taken at 0·55, and slip at 25 per cent., then I.H.P. $= AV^3 \div 807$.

The *effective diameter* of a radial wheel is usually taken from the centres of opposite floats; but it is very difficult to say what is absolutely that diameter, as much depends on the form of float, the amount of dip, and the waves set in motion by the wheel. The slip of a radial wheel is from 15 to 30 per cent., depending on the size of float.

The area of one float may be found by the following rule :—

$$\text{Area of one float in sq. ft.} = \frac{\text{I.H.P.}}{(D \times R)^3} \times C.$$

D is the effective diameter in feet, R is the revolutions per minute, and C is 1,400,000.

The breadth of the float is usually about $\frac{1}{4}$ its length, and its thickness about $\frac{1}{8}$ its breadth. The number of floats varies directly with the diameter, and there should be one radial float for every foot of diameter.

Example.—To find the particulars of the floats for a radial wheel 15 feet effective diameter, the I.H.P. of the engines being 400. Revs. 30 per min.

$$\text{Area of one float} = \frac{400 \times 1{,}400{,}000}{(15 \times 30)^3} = 6{\cdot}14 \text{ sq. ft.}$$

$$\text{Length} \times \frac{\text{Length}}{4} = 6{\cdot}14.$$

$$\text{or Length} = \sqrt{4 \times 6{\cdot}14} = 5 \text{ feet.}$$

$$\text{Breadth} = \frac{5}{4} = 1{\cdot}25 \text{ feet, or 15 inches.}$$

$$\text{Thickness} = \frac{15}{8}, \text{ or } 1\tfrac{7}{8} \text{ inches.}$$

Number of floats, 15.

The floats are usually made of elm, although any tough strong wood which withstands the action of water will do equally well. They are commonly rectangular in form, with the corners rounded off and the dipping edges bevelled at the back, so as to enter the water as easily as possible.

To avoid the shock of entry, the floats are sometimes shaped so that the middle of the float enters first, the sides being cut away taper for that purpose. To avoid the oblique action of the float on entry, the arms are sometimes bent so that the float is nearer to the perpendicular before arriving at the point immediately below the centre of the wheel; it is consequently more oblique on emerging than that of the common wheel, and tends to increase the lifting of the water then; it is also bad for going "astern."

The wheel is also sometimes made with the floats inclined to the axis, so as to throw a stream of water away from the hull of the ship when going ahead, and it is said that such wheels are more effective than the ordinary ones.

Feathering Floats.—It is easily seen that the larger the diameter of the wheel with radial floats, the less is the obliquity at entry and exit; but independently of the objection that a larger engine is required to drive the larger wheel with the same percentage of slip, there is always the practical difficulty of arranging for a wheel of large diameter without seriously interfering with the ship's design. By means of the feathering floats, all the advantages of a large wheel are obtained without the disadvantages.

The floats of a feathering wheel, instead of being bolted to the arms, are pivoted on them; that is, they are free to move on an axis parallel to the axis of the wheel; they are retained in position by means of levers attached to them, which are operated on by an eccentric-pin or sheave by means of rods. This eccentric is so set with respect to the centre of the wheel as to cause the float to be nearly vertical when entering and leaving the water. Fig. 116 shows an ordinary feathering wheel, from which it is seen that the floats are hung by gudgeons secured in the cross bars on their back, which fit in eyes in brackets forged with the paddle arms; one of the cross bars has a tail forged with it, which acts as the lever to turn the float. The eccentric-pin has a boss or strap running loose on it, and turned round by means of a radius rod, called the *king rod* L E (fig. 116), attached to one of the float levers; the other radius rods are jointed both at the levers and boss. The eccentric-pin is secured in its proper position to the sponson beam, so that when the wheel moves round each float is brought in turn to the proper position for entry, &c. The feathering is sometimes done by an eccentric on the inner side of the wheel attached to the bracket carrying the bearing, or to the bearing itself. This method suits small steamers, and avoids risk of damage from the sponsons getting crushed or bent on coming alongside ships or quays. The friction, however, is much greater than with the pin, &c.

To design a feathering wheel so that the floats shall enter edgeways when going at full speed, take P a point on the face of the float just entering the

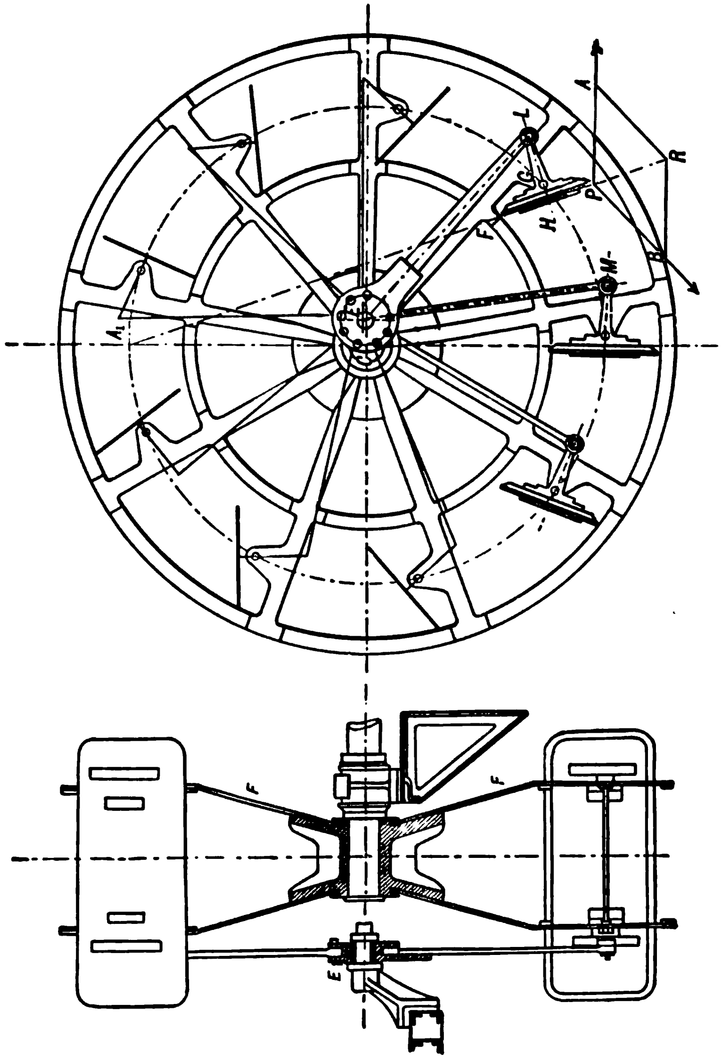


Fig. 116.—Paddle-wheel with Feathering Floats.

water, draw P A parallel to the water-line, and cut-off P A equal to the speed of the ship through the water; draw P B tangential to the circle through P, whose centre is the centre of the wheel; and cut-off P B equal to the speed of the wheel on that circle. Complete the parallelogram A P B; the diagonal

P R is the resultant of P A and P B, and is consequently the direction in which the float must move in that position to enter edgeways.

Produce R P, and draw parallel to it a line at a distance equal that of the centre of the gudgeon from the face of the float, cutting the circle of centre lines of gudgeons at G.

Draw G H at right angles to P F, and make H F and H P each equal half the breadth of the float. Make G L equal to the length of the lever required.

Now draw another float whose face is perpendicular and immediately under the centre of wheel, and the end of whose lever is M.

With centres M and L, and radius equal to G C, draw arcs of two circles intersecting at E. Then E is the centre of eccentric pin, and L E the length of the radius rods.

With E as centre and L E radius, draw a circle on which the centres of the lever ends of all the other floats lie.

It is usual to make the wheel not less than equivalent to a radial wheel of double the diameter, and to construct it by taking a float vertically under the centre of shaft, and G the point on which the next float pivots. With G as centre and G H radius, draw the arc of a circle, and from the point A vertically above the centre of the wheel draw A H a tangent to this arc; then the face of the float must lie on A H, and may be drawn as before. Likewise the centre of the eccentric may be found as before. With small wheels and high revolutions it is not sufficient to take the equivalent of double the diameter—in fact, in some cases even three times is not too much. But whatever equivalent is taken the same construction herein described holds good.

Diameter of a Feathering Wheel is found as follows:—The amount of slip varies from 12 to 20 per cent., although when the floats are small for fast running, it is as high as 25 per cent.; a well-designed wheel on a well-formed ship should not exceed 20 per cent. under ordinary circumstances.

Let v be the velocity of the ship, and s be the slip in feet *per second*, and R the number of revs. per minute at which it is intended to run the engine.

Then V the velocity of wheel at float centres $= v + s$.

D is the diameter of wheel at centres and $V = \pi D R$. Hence,

$$v + s = \frac{\pi D R}{60} \text{ and } D = \frac{v + s}{\pi R} \times 60.$$

Or if K is the speed of the ship in knots per hour, and S the percentage of slip, and R the revolutions per minute,

$$\text{Diameter of wheel at centres} = \frac{3222\ K}{(100 - S) R}.$$

Example.—To find the diameter of a wheel for a ship intended to steam 16 knots, with 40 revolutions of the engines and 20 per cent. slip.

$$\text{Diameter} = \frac{3222 \times 16}{80 \times 40} = 16{\cdot}1 \text{ feet.}$$

As before thrust $= 2\,A\,V\,(V - v)$. But in this case A is the area of a pair of floats for side wheel steamers; slip 20 per cent., and efficiency 0·7.

$$\text{Indicated horse-power} = \frac{2\,A\,V\,(V - v) \times 60\,v}{0{\cdot}7 \times 33{,}000} = \frac{A\,V^3}{1203}.$$

$$V = \frac{\pi D R}{60};\ V^3 = \frac{\text{I.H.P.} \times 1203}{A} = \left(\frac{\pi D R}{60}\right)^3.$$

$$\text{Then area of one float } \frac{A}{2} = \frac{\text{I.H.P.}}{(D \times R)^3} \times 3{,}815{,}000.$$

$$\text{Diameter of wheel} = \frac{197}{R} \sqrt[3]{\frac{\text{I.H.P.}}{A}}.$$

$$\text{The number of floats} = \frac{D + 2}{2}.$$

The breadth of the float = 0·35 × the length.
The thickness of floats = $\frac{1}{12}$ the breadth.
Diameter of gudgeons = thickness of float.

When a ship is working always in smooth water the immersion of the top edge should not exceed $\frac{1}{8}$ the breadth of the float; and for general service at sea an immersion of $\frac{1}{2}$ the breadth of the float is sufficient. If the ship is intended to carry cargo, the immersion when light need not be more than 2 or 3 inches, and should not be more than the breadth of float when at the deepest draught; indeed the efficiency of the wheel falls off rapidly with the immersion of the wheel, and for this reason the radial wheel was so long retained for sea-going cargo boats, as it permitted of the floats being reefed—that is, brought nearer to the centre.

Paddle-wheel Frames, &c.—The load on the wheel due to the resistance of a float is taken immediately by the pair of arms to which the float is attached, and by means of the rims is partially transmitted to the other arms. For ships working only in smooth water a much lighter wheel is sufficient than would do for those liable to the shocks of heavy seas. In any case the strength of the wheel must be sufficient to transmit the power of the engine to the water in the float-race, and as the twisting power exerted on the crank is approximately proportional to the cube of the diameter of the shaft inner journal, the strength of the wheel-frame must be in the same proportion. The strength of the arms of the wheel will vary directly as their thickness and as the square of their breadth. Hence, if D is the diameter of the *inner* or engine journal of the paddle-shaft, t the thickness of an arm whose breadth is b, and n the number of arms on one wheel (2 for each float),

$$n \times t \times b^2 \text{ varies as } D^3,$$

and

$$n \times t \times b^2 = K \times D^3,$$

or

$$t \times b^2 = \frac{K}{n} \times D^3.$$

When the shaft is of iron or mild steel K = 0·7 for a section near the boss, and 0·45 for a section near the inner rim when there are two rims. When the wheel has only one rim, so that the floats are attached to the projecting ends of the arms, the arms must be made much stronger, as the whole strain may come on the pair of arms, and cause a severe bending strain close to the inner rim.

The section at the arm outside the rim may be found in a similar way, on the assumption that the distance of float centre from the rim bears a constant ratio to the diameter of the wheel (about $\frac{1}{10}$); then, if t be the thickness and b the breadth as before,

$$t \times b^2 = K \times D^3.$$

If the shaft be of iron, K = 0·1.

Usually $\frac{b}{t} = 5$ for arms near the boss, and 3·5 to 4 for arms near the rim. When there are two rims, the arms are of uniform breadth and thickness between the rims.

When there is only one rim, $\frac{b}{t} = 6$ at the parts of the arm just at the outer edge of the rim.

With small wheels it is not convenient to have the same bracing that is found so necessary with large ones; the arms are therefore made heavier than given by the above rules when there are no bracing stays, and should be in accordance with the following:—

$$t \times b^2 = \frac{K}{n}(D^3 + 100).$$

The section of the inner rim is 0·8 that of the arms near to it, and $\frac{b}{t} = 5$.

The section of the outer rim is equal to that of the arm near it, and $\frac{b}{t} = 4$.

When there is only one rim its section is equal to that of the arm near the boss, and $\frac{b}{t} = 4$.

The inner rim of the wheel is stayed to the boss by round ties, whose diameter is about twice the thickness of the rim. They are bolted to the rim between the arms, and to the side of the boss opposite, so as to be diagonal with a pair of arms. This gives good support to the whole framework, and prevents racking and side-play.

Example.—To find the scantlings of a feathering wheel having inner and outer rims, the diameter of paddle-shaft journal at the engine being 10 inches, and the number of floats 8 to each wheel. The number of arms therefore is 16.

(1) Section of arms near boss, $\left(\frac{b}{t} = 5\right)$.

$$t \times b^2 = \frac{0 \cdot 7 \times 10^3}{16} = \frac{700}{16}.$$

Since $b = 5\,t$,

$$25\,t^3 = \frac{700}{16},$$

$$t = \sqrt[3]{1 \cdot 75} = 1 \cdot 2 \text{ inches,}$$

and

$$b = 5 \times 1 \cdot 2, \text{ or } 6 \text{ inches.}$$

(2) Section of arms near rims, $\frac{b}{t} = 4$.

$$t \times b^2 = \frac{0 \cdot 45 \times 10^3}{16} = \frac{450}{16}.$$

Since $b = 4\,t$,

$$16\,t^3 = \frac{450}{16}.$$

$$t = \sqrt[3]{1{\cdot}75}, \text{ or } 1{\cdot}2 \text{ inches},$$

and

$$b = 4 \times 1{\cdot}2, \text{ or } 4{\cdot}8 \text{ inches}.$$

(3) The section of inner rim = 0·8 (1·2 × 4), or 3·84 square inches, and since $\frac{b}{t} = 4$, the area $= 4\,t^2$. Hence

$$t^2 = \frac{3{\cdot}84}{4}; \quad \text{or } t = 0{\cdot}98 \text{ inch},$$

and

$$b = 4 \times 0{\cdot}98, \text{ or } 3{\cdot}92 \text{ inches}.$$

The outer rim has the same sectional area as the arms. The iron of arms would be then 6 inches by $1\frac{3}{16}$ inches at boss, and $4\frac{3}{4}$ inches by $1\frac{3}{16}$ inches at rims. The outer rim $4\frac{3}{4}$ inches by $1\frac{3}{16}$ inches, and the inner rim 4 inches by 1 inch, stayed diagonally by bars 2 inches diameter.

For the wheels of steamers running only in smooth or nearly smooth water, the values of K may be reduced by 15 per cent. in each case, and if it is necessary to make the wheel as light as possible, the strength of the various parts should be calculated by finding the value of R, the resistance at the floats, and assume that the whole power of the engine may be applied to one wheel.

Let T be the maximum twisting moment in inch-pounds, as formed by the rules in Chapter ix., D the diameter of the wheel to float centres in inches. Then

$$R = T \div \frac{D}{2}.$$

$R \times \frac{D}{2}$ is the bending moment at the centre of the wheel; and $R \times \frac{D}{4}$ that half way between the centre and float-centres.

The bending moment at the middle of the arms is therefore

$$\frac{2\,T}{D} \times \frac{D}{4} = \frac{T}{2}.$$

And if n be number of arms, or twice the number of floats,

$$\text{Bending moment on one arm} = \frac{T}{2\,n}.$$

Then

$$t \times b^2 = \frac{T}{F \times n} \times 3.$$

F may be taken at 7500 for ordinary, and 8500 for really good iron or steel.

The gudgeons on which the floats of a feathering wheel are hinged are not, of necessity, placed at their centre line horizontally, and it is a very general custom so to fix them that the larger part is above the axis, so that

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when the float is entering the water and leaving it, there is not so much load on the radius rods. As a rule, the line of gudgeons should be $\frac{3}{8}$ of the breadth of the float from the edge nearest the centre of the wheel.

The gudgeons and pins of the feathering gear are of iron, cased with brass, and the holes in which they work should be bushed with lignum vitæ. If the ship usually works in sandy water, the pins may be of iron, and the holes bushed with white metal, which withstands the cutting action of the sand particles better than do the brass and wood.

The eccentric-pin on the sponson beam should also be cased with brass, and the boss to which the feathering rods are attached should be bushed with lignum vitæ, as water is the only lubricant applied to them.

The outer bearing of the paddle-shaft should be twice the diameter of the shaft in length, very strongly made, and firmly secured to the bracket on the ship's side, as the whole of the thrust, besides the weight of the wheel, is taken on it. The force acting on it is the resultant of the thrust and weight, whose direction is diagonal, and its magnitude easily calculated. The bearing must be so formed as to take the load, and to admit of adjustment in that direction. It is always downward, so that the caps of the bearing need not be very strong. The lubricant is to a great extent water, but an arrangement should be made for oiling, and a cavity left in the cap for a piece of tallow, or a mixture of tallow and soft soap. The shaft should have fitted to it a stuffing-box and gland, where it passes through the skin of the ship.

Screw Propellers.—The simple screw, as first fitted, consisted of a part of a true helix, cut off by two parallel planes perpendicular to the axis, and it had therefore only one blade, strictly speaking. By cutting a double helix, or a double-threaded screw in this way, two blades are obtained; and in this form, with very slight modifications, the screw propeller was used for many years. It was found, however, that by cutting away the corners, especially those on the forward or leading edges, the vibration was very much reduced, and in course of time the paring process left the screw blade very much as it is generally found now.

The number of blades was, however, increased from two to as many as six, and as there was no scientific or even practical test made of the relative advantages of six or two blades, it remained for a happy accident to the six-bladed propeller of a canal boat to demonstrate that six blades were too many; since then, ample proof has been given that, although in smooth water their efficiency is very high, two blades are not sufficient in a rough sea; and that if one of three blades be broken off, the propeller is so badly balanced as to severely strain the engines, so that four blades has become the rule for the propellers of sea-going ships of the mercantile marine, as also for the propellers of battleships and very large cruisers.

For smooth water the two-bladed propeller is the most efficient, but its efficiency is rapidly impaired so soon as the ship begins to pitch. The efficiency of a screw-propeller depends on so many things, some of which are external, that no rules can be laid down which will command the respect of practical engineers. A screw which will work with most satisfactory results on one ship, may be most inefficient on another, although driven by similar engines, and at the same number of revolutions; the reason for such difference, however, is not far to seek when properly looked for.

Although very much has been written on the subject, and many men of undoubted ability have spent time and money in making researches, still the best informed are liable to make the most egregious mistakes in designing a screw, as evidenced on the first trials of more than one most important ship.

That opinions differ very widely on the subject may be seen by referring to the Patent Records, as well as to the Transactions of the various Scientific Institutes.

Professor Rankine (Appendix to *Steam Engine*) states—"The efficiency of the propeller is $\frac{V}{V + S}$, V being the velocity of the ship in feet per second, and S the *slip* of the propeller in feet." In other words, the less the slip, the higher the efficiency; this, however, is only true theoretically, on the supposition of there being no friction. The late Dr. Froude, than whom no one has given more patient attention to this most important subject, says that, . . . "instead of its being correct to regard a large amount of slip as a proof of waste of power, the opposite conclusion is the true one. To assert that a screw works with unusually little slip, is to give a proof that it is working with a large waste of power,"* which is the exact contrary of the conclusion arrived at from Rankine's investigation, but is in accordance with everyday observation on the performances of ships. It must not, however, be supposed that a screw having a large amount of slip is necessarily an efficient one, for, as has already been shown, the slip varies inversely as the area of transverse section of the stream for a constant power.

A screw works in water disturbed more or less by the passage of the ship, and is liable to be very seriously interfered with by the "wake" or water following the ship, and therefore cannot be dealt with as if working in still water.

Let P be the pitch of the screw, R the number of revolutions per second; then,

$$\text{Velocity of stream from it } V = P \times R.$$

Let v be the velocity of the ship in feet per second, A the area of section of stream from the screw, which is approximately the area of a circle of the same diameter, less the area of transverse section of the boss. Then,

$$V - v \text{ is the slip (\textit{vide} p. 309).}$$

As before explained, A × V is the volume of water projected *from the ship* in cubic feet per second, and taking the weight of a cubic foot of *sea*-water at 64 lbs., and the force of gravity at 32,

$$\text{Momentum of stream} = \frac{A \times V \times 64}{32} (V - v) = 2\, A \times V\, (V - v).$$

This is the thrust of the screw on the ship exerted along the shaft in pounds.

Professor Rankine gives (*Rules and Tables*, p. 275) this same rule in another and more convenient form for practical use, viz.:—

RULE V.—*To calculate the thrust of a propelling instrument (jet, paddle, or screw) in pounds: multiply together the transverse sectional area, in square feet, of the stream driven astern by the propeller; the speed of the stream relatively to the ship in knots; the real slip, or part of that speed which is impressed on that stream by the propeller, also in knots; and the constant* 5·66 *for sea-water, or* 5·5 *for fresh water.* That is, if S is the speed of the screw

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* *Transactions of the Institution of Naval Architects*, Vol. xix., 1878.

TABLE XXII.—PARTICULARS OF SCREW PROPELLERS.

SOLID CAST IRON.

Ship. S = Single Screw. T = Twin Screw.	Diam.	Pitch.	Blades.				Length of Boss.	Diam. of Shaft.	I.H.P. / R	Material.		Weight.
			No.	Max. Breadth.	Thick.	Total Surface.				Boss.	Blades.	
	Ft. Ins.	Ft. Ins.		Ft. Ins.	Ins.	Sq. ft.	Ft. Ins.	Ins.				Cwts
Dn., S.,	19 6	26 6	4	3 10½	10	112	3 9	17¾	64·7	...	...	250·5
Nno., S.,	18 3	26 0	4	3 9	9	96	3 0	13	36·1	...	...	202 5
Slr., S.,	16 8	23 6	4	4 3	8½	92	3 6	13¾	36·7	...	...	189·0
Ddo., S.,	17 6	17 6	4	3 6	7½	90	2 9	13⅛	34·2	...	...	124·7
Ard., S.,	16 0	20 0	4	3 9½	7⅛	85	2 9	13¾	33·0	...	...	1[illegible]1·0
Arl., S.,	16 0	18 0	4	2 9¼	7¼	74	2 9	12¼	21·8	...	...	110·75
Eld., S.,	13 6	21 0	4	3 3¾	7¼	68	2 9	11⅛	28·8	...	...	98·90
Hyd., S.,	12 9	16 0	4	3 3½	6⅛	58	2 5	11	14·8	...	...	75·65
Esw., S.,	13 0	16 4	4	2 5½	5¾	48	2 3	9¾	10·3	...	...	61·40
Alt., S.,	10 3	17 3	4	2 11¼	5	43	2 1	8⅞	8·7	...	...	44·10
Bln., T.,	12 0	16 6	3	3 3	8	40·5	2 6	11¾	20·0	...	...	75·50
Qnw., S.,	9 9	13 6	4	2 4	4	33·5	1 9½	6⅞	6·10	...	...	29·60
Hwh., T.,	9 0	14 0	3	2 10¾	4¾	26·0	1 10	7¾	5·69	...	...	27·50
Erp., S.,	9 0	13 0	3	2 2½	4⅛	21·0	1 6	6⅞	4·00	...	...	20 30
Wtm., S.,	7 8	9 0	3	2 1½	3⅛	17·5	1 3	5⅛	2·30	...	...	12·80
Skb., S.,	6 3	7 0	3	1 9¼	3¼	12·5	1 1⅛	4¾	1·15	...	...	8·50
Fbc., S.,	4 6	5 0	3	1 1¾	2⅛	5·5	0 9	3⅝	0·42	...	...	2·90

LOOSE BLADES BOLTED ON.

Ship.	Diam.	Pitch.	No.	Max. Breadth.	Thick.	Total Surface.	Length of Boss.	Diam. of Shaft.	I.H.P. / R	Boss.	Blades.	Weight.
	Ft. Ins.	Ft. Ins.		Ft. Ins.	Ins.	Sq. ft.	Ft. Ins.	Ins.				
Cpt., S.,	20 3	25 0	4	4 5¼	8½	120	3 9	18¼	70·0	C. steel	M. bronze	311·5
Mtl., S.,	19 0	22 0	4	3 8½	6	96·5	3 6	15⅛	40·0	C. steel	Steel	196·7
Str., S.,	19 0	23 0	4	4 4½	8¼	96·0	3 7½	15⅛	41·2	C. iron	C. iron	288·0
Czg., S.,	17 0	21 6	4	4 2	9¼	90·0	3 6	15	41·0	C. iron	C. iron	223·9
Ldn., T.,	17 6	19 9	4	4 0	7¼	86·0	4 0	17½	72·0	M. bronze	M. bronze	261·0
Std., T.,	16 0	25 0	3	3 1¾	6½	77·0	3 9	18	75·0	D. metal	D. metal	216·5
Stm., S.,	17 0	21 0	4	3 5¾	7⅜	72·0	2 8	13⅛	30·1	C. iron	C. iron	183·7
Rmo., S.,	15 0	23 0	4	3 8¾	6¾	63·6	2 9	13½	25·0	C. iron	C. iron	170·3
End., T.,	16 6	22 0	3	3 10	7¼	60·0	3 4½	16⅛	54·5	A. bronze	A. bronze	166·6
Kvn., S.,	14 0	17 0	4	2 9	6	55·0	2 1	9¾	11·2	C. iron	C. iron	99 5
Apl., T.,	13 0	17 0	3	3 3½	6¼	40·0	3 0	14⅛	32·9	A. bronze	A. bronze	107·7
Clm., T.,	12 0	16 0	3	3 2¼	5	39·0	2 6	11¾	20·1	C. steel	S. bronze	71·80
Cbg., T.,	11 6	18 6	3	3 1¼	4¼	34·0	2 2	10¼	13·3	C. iron	S. bronze	62·85
Prl., T..	12 6	14 0	3	2 9	6	33·0	2 6	12⅛	22·8	A. bronze	A. bronze	76·5
Prs., T.,	10 6	12 0	3	2 5¼	4⅞	24·0	2 6	12⅝	16·3	A. bronze	S. bronze	64·7
Blc., T.,	9 3	10 0	3	2 5¼	3¾	21·3	2 1⅛	9	7·5	A. bronze	A. bronze	36·0
Slm., T.,	6 7	8 9	3	1 10¾	2	12·7	1 0¾	6⅛	5·0	S. bronze	S. bronze	7·45

in knots, s the speed of the ship in knots, A the area of the stream in square feet (of sea-water),

$$\text{Thrust in pounds} = A \times S(S - s) \times 5{\cdot}66, \text{ or } D^2 \times S(S - s) \times 4{\cdot}45.$$

The effective slip or the actual mean velocity imparted to the water cannot be found accurately any more than can the mass of water set in motion. The effective thrust, however, may be found approximately, thus—

Rule.— $$\text{Effective thrust} = (D \times S)^2 \div F.$$

D the diameter in feet, S the speed of screw in feet per second, and F varies from 3·3 with small quick propellers to 5 with large slow ones. Ordinary merchant steamers, 5 to 4·5; fast-running merchant and naval ships, 4·0; very high speed, as with turbines, 3·3.

Apparent Slip.—The difference between the speed of the ship past the water which is removed from the hull some few yards, and the speed of the screw calculated by multiplying the pitch by revolutions, is called the *apparent slip*, as it is not always the measure of the *real slip*, which is the velocity imparted to the water by the screw in a direction opposite to that of the ship. No propeller can propel a ship without *real slip*, although it is a common occurrence to find that the screw has no apparent slip, and sometimes has even negative slip, so that the ship seems to be going faster than the screw.

Negative Slip.—When the phenomenon of negative apparent slip was first observed, it was thought by some to be due to false measurement of the effective pitch, and this theory was to some extent strengthened by the fact that it often occurred with screws having a variable pitch. More thoughtful observers traced it to another cause.

It was observed that every ship dragged, so to speak, a casing of water with it, and that there was with some ships a following stream just at the stern, which seemed to be always running into the space left by the ship, so that the velocity of the ship with respect to that casing and following stream is less than the real velocity past the still water—in other words, the velocity of the ship through the water immediately surrounding it is less than the real velocity v. The screw works in this disturbed water; consequently, to calculate its *real slip*, the forward velocity of the following stream must be known.

There is, however, much yet to be learnt before the exact causes of negative slip are fully understood, for, although some ships from their form, or some such peculiarity, are more prone than others to cause negative slip, yet a propeller can be fitted which will produce positive slip, and that without loss of propelling effect, but on the contrary it is frequently done with positive gain. As an example of this, the case of H.M.S. "Amazon" may be cited. With her original screw, which was a 4-bladed Mangin 15 feet diameter and 12 feet 6 inches pitch, a speed of 12·079 knots only was obtained with 1940 I.H.P., the slip being 14 per cent. negative; while with a two-bladed Griffith's screw 15 feet ¾ inch diameter and 13 feet 9 inches pitch, a speed of 12·396 knots was obtained with only 1664 I.H.P., the slip being then 3·16 per cent. positive.

If a propeller gives negative slip, or only no slip, it should be changed at the first opportunity, as all experience has shown that it is not working efficiently when such is the case.

Diameter of Screw.—The size of a screw depends on so many things, that it is very difficult to lay down any hard rule for guidance, and much must always be left to the experience of the designer so to allow for all the circumstances of each particular case as to give the proper proportions.

If a screw blade breaks through the surface of the water, it will carry down with it a certain amount of air, which very much decreases the efficiency, and causes sudden accelerations of the speed of the screw, the mixture of air and water being of lower density than that of water alone. This acceleration is called *racing*, and may always be observed when the propeller is not wholly immersed, or when the ship pitches even very slightly. For this reason a small propeller wholly immersed is better and more efficient than a larger one whose top edge is above or even quite close to the water surface. Hence, the diameter of the propeller of ordinary ships should be always less than the draught of water. The diameter of the screw must also bear some relation to the turning power of the engine, for if made of too small diameter it cannot absorb the power of the engine, however coarse the pitch may be. Some account also has to be taken of the form of the ship; if the stern lines are bluff, as in most cargo steamers, propellers of larger diameter are necessary.

It has been shown by the late Dr. Froude, in the paper already alluded to, that if a blade moved through the water without surface friction and "head" resistance, the larger the diameter, the better; also, that if the efficiency of the mechanism was the same at any number of revolutions, a fine pitch propeller might be used with advantage, as being more efficient than a coarse pitch one. But since the efficiency of some engines is much higher at moderate than at the maximum rates of speed, the total efficiency of the screw and engine is often improved by increasing the pitch of the screw. Further, since there is surface or skin friction, and it increases per square foot of surface as the square of the distance from the centre, and the distance at which it acts also increases with the distance from the centre, the fractional resistance per square foot varies as the cube of the distance from the centre, so that any increase of diameter means large increase in resistance. For these reasons small screws with coarse pitch often give far better results than large ones of fine pitch on the same ships; this was proved on the trials of H.M.S. "Iris," and on many other occasions.

The magnitude of the skin resistance of a propeller is well illustrated by Dr. Froude, who says—

"Take the case of a screw 20 feet diameter, making 80 revolutions per minute; the tips of the blades are travelling at a speed of about 50 knots; now, the resistance of a surface so short in the line of motion as a screw blade, *even when its surface is quite smooth*, is as much as 1¼ lbs. per foot at 10 knots, and is nearly as the square of the speed, and as each square foot of blade area involves 2 square feet of skin, the resistance of each is over 60 lbs.; thus, making some allowance for thickness and bluntness, there is involved in driving it at 50 knots at least 10 I.H.P., and collectively the outmost foot of four such blades, each 3 feet wide, would absorb fully 120 I.H.P. in surface friction; and though the parts nearer to the root move with proportionally less speed, and therefore with less resistance, yet, on the other hand, screw blades are generally rough from the sand, and have probably a still higher coefficient of frictional resistance."

If a screw is of sufficient diameter, and has ample blade area, any addition beyond the tip will act then quite as a brake, and seriously reduce its efficiency.

By referring to the diagram (fig. 116*a*) the pressure in pounds per square inch of any part of a propeller blade can be ascertained, and this permits of an easy calculation of the strength of a propeller blade as well as the resistance of the propeller itself when working under various conditions. The diagram has been made originally by Messrs. J. Stone & Co., brass founders and makers of bronze propellers.

Thrust of a Screw Propeller.—The thrust of a propeller has been shown to vary directly as the area of the disc—that is, it varies as the square of the diameter; it has been also shown that the thrust varies as the square of the velocity of flow—that is, as the square of the product of the revolutions and pitch. But the speed of the ship varies as the product of the revolutions and pitch, and the effective horse-power should vary as the product of thrust and speed.

The actual slip of a screw propeller, or that amount of motion imparted to the column of water projected by it, cannot be calculated with any degree of accuracy, as it probably varies considerably from the boss to the tip, but there is good reason to suppose that the mean effective acceleration is greater than the apparent slip, and, for purposes of calculation, may be assumed to be 12½ per cent. of the pitch multiplied by revolutions in the case of screws of large diameter, as obtain in the mercantile marine and formerly in the Navy; with the smaller and faster running ones, 15 per cent.; and with the high-revolution reciprocating engines and turbines, 20 per cent. Twin screws, being in a better position for efficiency, may be accorded credit for 2 to 3 per cent. over single screws.

Let d be the diameter, p the pitch in feet, R the revolutions per minute, and s the advance in feet per second, x a fraction of s and equal to the effective slip or velocity imparted to the water, g the gravity equal to 32, and the weight of a cubic foot of sea water 64 lbs. Then—

$$\text{Thrust in pounds} = \frac{\pi d^2}{4} \times s \times \frac{64}{g} \times xs = d^2 \times s^2 \times x \times 1{\cdot}6.$$

Hence,

$$\text{Diameter of screw} = \sqrt{\frac{\text{Thrust}}{s^2 \times x \times 1{\cdot}6}}.$$

Now, $\text{Thrust is approximately} = 0{\cdot}8 \times \text{indicated thrust} = \dfrac{0{\cdot}8 \times \text{I.H.P.} \times 33{,}000}{60 \times s}.$

Substituting this in the above. Then—

$$\text{Diameter of screw} = 16{\cdot}6\sqrt{\frac{\text{I.H.P.}}{s^3 \times x}}.$$

If the slip is 12½ per cent., $x = \frac{1}{8}$. Then, for ordinary merchant single-screw ships.

Rule.—$\text{Diameter of screw} = 50\sqrt{\dfrac{\text{I.H.P.}}{s^3}}.$

For very high-speed screws with an effective slip of 20 per cent., $x = \frac{1}{5}$. Then—

$$\text{Diameter of screw} = 37\sqrt{\frac{\text{I.H.P.}}{s^3}}.$$

But the diameter in practice is governed by other factors than have been taken into account in the above. It is well known that a bluff cargo boat requires a larger screw than does a ship of the same dimensions and power, but of finer lines, and that the finer the lines are the smaller diameter may the screw have.

The following rule may be observed, therefore, in determining the diameter of screws for every kind of ship to give the best results in smooth water at or near the full speed. If a ship is frequently exposed to a rough sea, or is to be on service with the screws partially immersed, a somewhat larger screw may be found of advantage.

***Rule.*—Diameter of screw** $= \text{F} \times \text{K}\sqrt[3]{\dfrac{\text{I.H.P.}}{\text{R}}}.$ F being the prismatic coefficient of fineness, and K, for a single screw, 7·2; for a twin screw, 6·5; for triple screws, 6·0 for the centre screw and 5·5 for the wings; and for quadruple screws, 5·35.

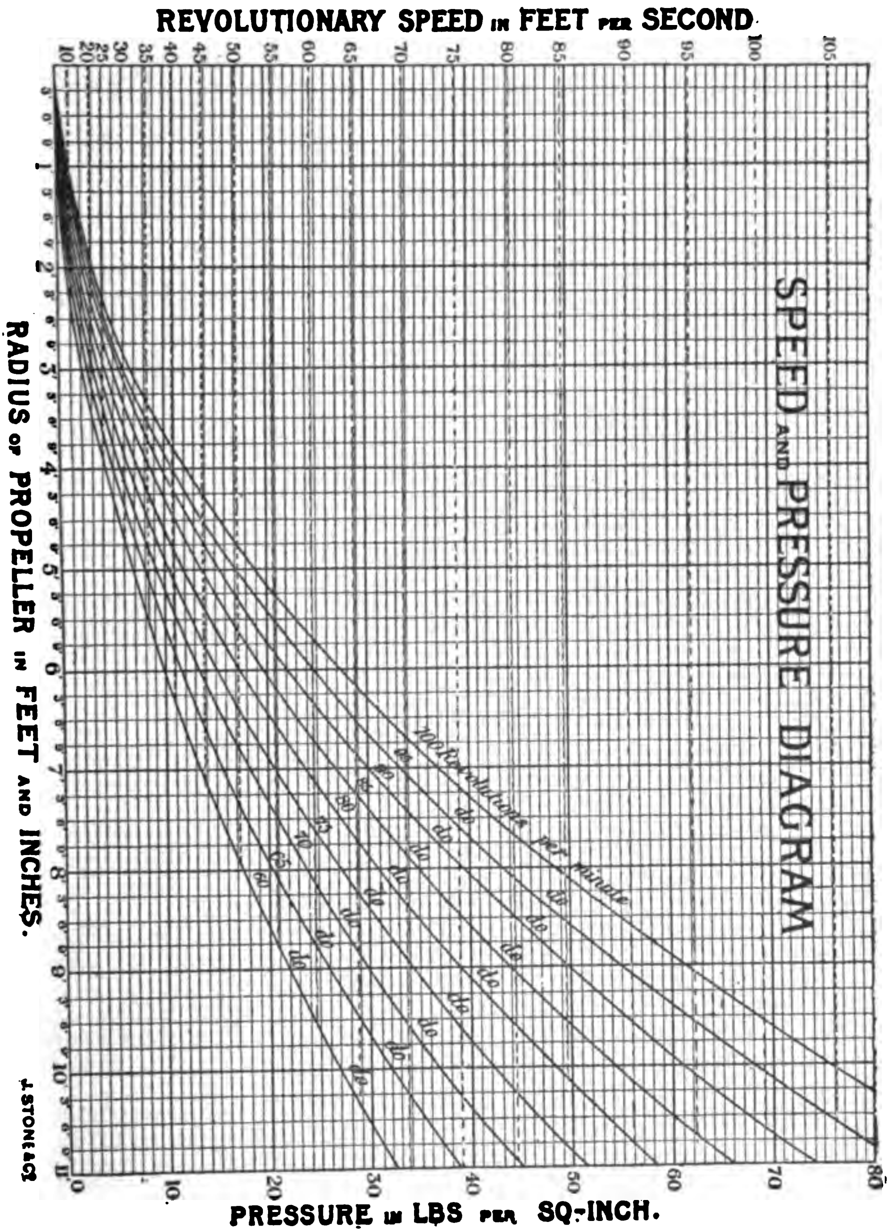
Example.—To find the diameter for a cross-channel twin-screw ship of 8000 I.H.P., 150 revolutions, coefficient of fineness 0·60, pitch of screw 15 feet, effective slip $\frac{1}{6}$.

By first rule—

$$d = 16{\cdot}6\sqrt{\frac{4000 \times 6}{37{\cdot}5^3}} = 11{\cdot}2.$$

By second rule—

$$d = 6{\cdot}5 \times 0{\cdot}6\sqrt[3]{\frac{4000}{150}} = 11{\cdot}6.$$



J. STONE & CO.

Fig. 116a.

Example.—To find the diameter of a screw for a fine-lined single-screw ship of 1200 horse-power when working at 70 revolutions, and to have a pitch of 20 feet.

$$\text{Diameter of screw} = 21{,}000\sqrt{\frac{1200}{(20 \times 70)^3}} = \frac{21{,}000}{1512}, \text{ or } 13\cdot89 \text{ feet.}$$

Example.—To find the diameter of a screw suitable for an engine of 250 N.H.P., for a cargo steamer, and having a pitch of 20 feet.

$$\text{Diameter} = \sqrt{\frac{250}{20} \times 20}, \text{ or } 15\cdot8 \text{ feet.}$$

Example.—To find the diameter of the single screw for a very fine steam launch whose engines develop 200 I.H.P. when working at 300 revolutions, the pitch to be 6 feet.

$$\text{Diameter of screw} = 25{,}000\sqrt{\frac{200}{(6 \times 300)^3}} = \frac{25{,}000}{5400}, \text{ or } 4\cdot6 \text{ feet.}$$

The blades of the large screws as usually fitted in full-cargo steamers reach out into the comparatively undisturbed water, while the small screws have a strong tendency to race, from the occasional failure of the water to flow in behind the buttock lines.

A screw steamer may have a full "entrance," and steam very fairly well if the "run is clean," while, on the other hand, a fine entrance will not compensate for a full "run."

Rules are often given for the diameter of a propeller without regard to the size or power of the engines, being generally based on the relation between the area of the screw's disc and that of the immersed midship section. It was also a very common thing to find in specifications "to fit as large a propeller as the draught of water will permit;" this means, in most cases, a propeller working inefficiently on account of the blades emerging from the water and carrying air down with them. If engineers generally would separate the efficiency of the screw from that of the machinery, such large screws would often be discarded.

Indicated Thrust.—The efficiency of the screw may be examined on the system proposed by the late Dr. Froude—viz., by constructing curves of *indicated thrust* on the same principle as that described in Chap. iii., for curves of I.H.P.

In this case, however, the ordinates represent the thrust as calculated from the I.H.P., or from the pressure on the pistons, and this for convenience is generally expressed in tons.

It is assumed that the pressure on the pistons multiplied by twice their stroke is equal to the thrust multiplied by the pitch, and if there were no loss by friction of machinery, &c., by the principle of work this would be true.

Let p be the mean effective referred pressure on the low-pressure pistons in pounds per square inch, n their number, A their area in square inches, L the length of stroke in feet, and P the pitch of the screw in feet; then,

$$\text{Thrust} \times \text{P} = p \times \text{A} \times n \times 2\,\text{L},$$

or

$$\text{Thrust} = \frac{p \times \text{A} \times n \times 2\,\text{L}}{\text{P}}.$$

If both numerator and denominator of the fraction be multiplied by R, the number of revolutions per minute,

$$\text{Thrust} = \frac{p \times A \times n \times 2\,L \times R}{P \times R};$$

but

$$\frac{p \times A \times n \times 2\,L \times R}{33{,}000} = \text{I.H.P.}$$

Therefore,

$$\text{Thrust} = \frac{\text{I.H.P.} \times 33{,}000}{P \times R}.$$

This is called the *indicated thrust*, and it was by constructing a curve of *indicated thrust* that the inefficiency of the original screws of H.M.S. "Iris" was discovered.

Dr. Froude* also explained the method by which he estimated the "*initial friction*," or "the equivalent of friction of the engines due to the working load." He said—"When decomposed into its constituent parts indicated thrust is resolved into several elements, which must be enumerated and kept in view. These elements are—1, the useful thrust, or ship's true resistance; 2, the augment of resistance, which is due to the diminution which the action of the propeller creates in the pressure of the water against the stern end of the ship; 3, the equivalent of the friction of the screw blades in their edgeway motion through the water; 4, the equivalent of the friction due to the dead weight of the working parts, piston packings, and the like, which constitute the initial or slow-speed friction of the engine; 5, the equivalent of friction of the engines due to the working load; 6, the equivalent of air-pump and feed-pump duty.

"It is probable that 2, 3, and 4 of the above list are all very nearly proportional to the useful thrust; 6 is probably nearly proportional to the square of the number of revolutions, and thus, at least at the lower speeds, approximately to the useful thrust; 5 probably remains constant at all speeds, and for convenience it may be regarded as constant, though perhaps in strict truth it should be termed 'initial friction.' If, then, we could separate the quasi-constant friction from the indicated thrust throughout, the remainder would be approximately proportional to the ship's true resistance.

"Now, on drawing a curve (of indicated thrust) . . . it becomes at once manifest in every case, that at its low-speed end the curve refuses to descend to the thrust zero, but tends towards a point representing a considerable amount of thrust, and it is impossible to doubt that this apparent thrust at the zero of speed, when there can be no real thrust, is the equivalent of what I have termed initial friction; so that if we could determine correctly the point at which the curve, if prolonged to the speed zero, would intersect the axis O Y (fig. 117), and if we were to draw a line through the intersection parallel to the base, the height which would be thus cut off from the thrust ordinates would represent the deduction to be made from them in respect of constant or initial friction, and the remainders of the ordinates between this new base and the curve would be approximately proportional to the ship's true resistance."

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* *Transactions of the Institution of Naval Architects*, vol. xvii., 1876.

Dr. Froude then explains his method, which is substantially as follows:—

Let O B, O C, O D represent the three progressive speeds, observed in the usual way, and B E, C F, D G the indicated thrust, calculated from the data observed at those speeds; through the points G, F, E draw the curve of indicated thrust. Let O A represent a low speed, which should not exceed 5 knots, and A H the corresponding indicated thrust; continue the curve to H, and at H draw a tangent to the curve cutting O Y. Take a point M between O A, so that $\frac{\text{O M}}{\text{M A}} = \frac{0{\cdot}87}{1{\cdot}0}$. Draw M K parallel to A H, and cutting the tangent line at K. Through K draw a line K T parallel to O D, cutting O Y at T. Then O T is the part cut off O Y by the curve, and represents the initial friction.

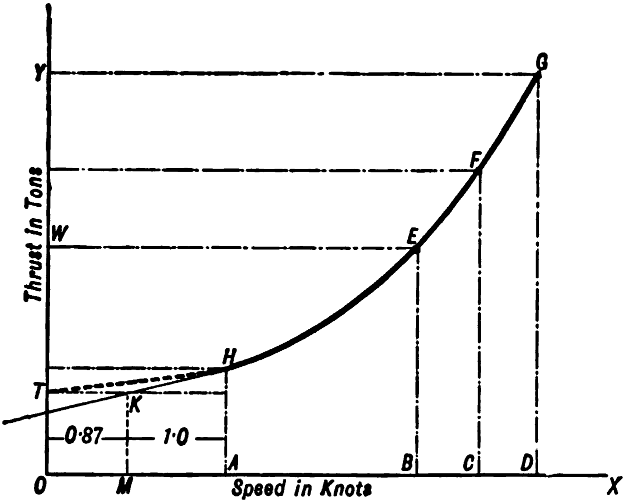


Fig. 117.—Curve of Indicated Thrust.

Dr Froude deduced, from careful investigation, that only 37 to 40 per cent. of the whole power delivered is usefully employed, and that "the constant friction is equivalent to from one-eighth to one-sixth of the gross load on the engine when working at its maximum speed and power."

Experiments with engines employed in generating electrical energy, and which run at constant revolutions whatever the load may be, require very little more power to overcome their own resistance when running at full load than with no load. The Belliss & Morcom engine, with forced lubrication, requires only 6 to 7 per cent. of its full power to do so; that is, its efficiency is 93 to 94 per cent. at full power, and zero when reduced to 7 per cent. of its power.

A marine engine does not work under these conditions, for its load varies with the cube of the revolutions. Its resistance per revolution, however, is not very much more at full speed than it is at slow speed. Hence the power to overcome its own resistance may be taken as $f \times \text{R}$; its gross power, however, is $\text{F} \times \text{R}^3$. Hence the useful work is $\text{F} \times \text{R}^3 - f\text{R}$, and its efficiency $\frac{F \times R^3 - f R}{F \times R^3} = \frac{F \times R^2 - f}{F \times R^2}$. But one of the causes of loss of efficiency in a marine installation is the friction, &c., of the propeller, and this varies as the square of the revolutions and the power as the cube, and makes a further complication.

Also, taking the power due to the ship's net resistance as E.H.P., and the six elements already enumerated when multiplied by

$$\frac{\text{Speed of ship in feet per minute}}{33{,}000} \text{ as S.H.P.}$$

Then

$$\text{S.H.P.} = 2{\cdot}347 \text{ E.H.P.}$$

Pitch of Screw.—Let S be the speed of the ship in knots, R the number of revolutions per minute, and s the slip in knots; then

$$\text{Pitch} = \frac{(S + s) \times 6080}{60 \times R}.$$

If the slip is expressed as so much per cent. of *the speed of screw*, which is the common way, and is x per cent. Then

$$S = \text{speed of screw} \left(1 - \frac{x}{100}\right),$$

or

$$\text{Speed of screw} = S \div \left(1 - \frac{x}{100}\right) = S \times \frac{100}{100 - x}.$$

Then

$$\text{Pitch} = \frac{S \times 100 \times 6080}{60\,R\,(100 - x)} = \frac{S}{R} \times \frac{10{,}133}{100 - x}.$$

Example.—To find the pitch of the screw for a ship whose speed is 15 knots, the slip is 10 per cent., and the number of revolutions 60 per minute.

$$\text{Pitch} = \frac{15}{60} \times \frac{10{,}133}{100 - 10}, \text{ or } 28{\cdot}14 \text{ feet.}$$

The slip of a well-proportioned screw on a ship of fairly good form should not exceed 8 per cent. when at *sea* full speed, and 10 per cent. when on *trial* full speed. If the slip is less than 5 per cent. at full speed, the screw is not of such proportions as to suit the particular ship. The large propellers of bluff cargo boats already alluded to, however, seldom exceed 5 per cent., and to them the remark does not apply. An excessive amount of slip does not of necessity imply waste of power, as it may arise from smallness of diameter; and also when a screw of larger diameter gives a better speed than that obtained by the original small one, it is often due to the increased efficiency of machinery at a reduced number of revolutions.

An abnormally large amount of slip may, however, be due to want of area of screw blade, and when this is the case the curve of indicated thrust exposes it, as it will then exhibit want of augmentation of thrust at the higher speeds.

Area of Screw Blade.—The best area of surface of screw blade is not easily determined, except by actual experiment, or from data derived from the performances of similar ships with similar screws. The general indication of too little surface is an abnormal amount of slip at the higher speeds with a low indicated thrust; a too large amount of surface generally causes the screw to show apparent negative slip, especially at the lower speeds; negative slip occurs more especially when there is extra surface of blade near the tips. In H.M. Navy negative slip was usually observed with four-bladed propellers, and especially with those of the "Mangin" type.

The area of blades given by the following rule is such as is generally found to give good results, and may be used by those who have no good experience to guide them :—

$$\text{Total area of screw blades} = \text{K} \sqrt{\frac{\text{I.H.P.}}{\text{revolutions}}}.$$

K for 4-bladed on	bluff line ships, -	16 for single screw;	12 for twin screws.
" "	fine " -	14·5 "	10·5 "
" 3- "	" " -	13 "	9·0 "
" "	very fine line ships,	12 "	7·9 "
" "	torpedo boats, &c.,	10·5 "	6·0 "

The projected area, or the area of the blade as drawn on an end elevation, such as would be seen on looking toward the stern of the ship, is often taken as the criterion of actual blade surface; and is to some extent a true one, since a coarse pitch blade requires more surface than a fine pitch one for the same power developed by the engine, and the *projected* area of the fine pitch blade will not differ much from that of the coarse pitch one.

Thickness of Blade.—The strength of a blade at the root, or part near the boss, will vary nearly directly as the breadth and as the square of the thickness, and for practical purposes may be assumed to do so. The strain on the propeller will vary as the cube of the diameter of the tunnel shafts, and very nearly as the cube of the diameter of the propeller shaft. In similar engines, the twisting power of the engines has been shown to vary as I.H.P. ÷ revolutions.

Let d be the diameter of the screw shaft close to the propeller, b the breadth of blade in inches at a distance of $1\frac{1}{2} \times d$ from the centre, and n the number of blades; R the number of revolutions per minute. Then

$$\text{Thickness of blade (at } 1\tfrac{1}{2} \times d \text{ from centre)} = \sqrt{\frac{d^3}{n \times b}} \times \text{K},$$

also

$$\text{Thickness of blade} \quad " \quad " \quad = \sqrt{\frac{\text{I.H.P.} \times f}{\text{R} \times n \times b}} \times \text{K};$$

f may be taken as 100 for compound engines of the ordinary two-cylinder type; 90 for three-cylinder compound, four-crank triple, and quadruple engines; and 85 for three-crank triple-compound engines. The value of K is 4 for cast iron, 2 for ordinary gun-metal, 1·6 for cast steel, and 1·5 for forged steel and bronzes of superior make. For ships engaged in the North Atlantic or other seas where rough weather largely prevails, these factors may with advantage be increased by 20 to 25 per cent. This is especially necessary for cargo vessels, which may often have to run "light."

The thickness of metal at the tip should be 0·2 of that at the root.

Propellers made of steel and bronze are very superior in strength to those of cast iron, even if made in accordance with these rules; for if made of the *same* strength only, the blades would not be *stiff* enough, and would vibrate very considerably, especially when racing.

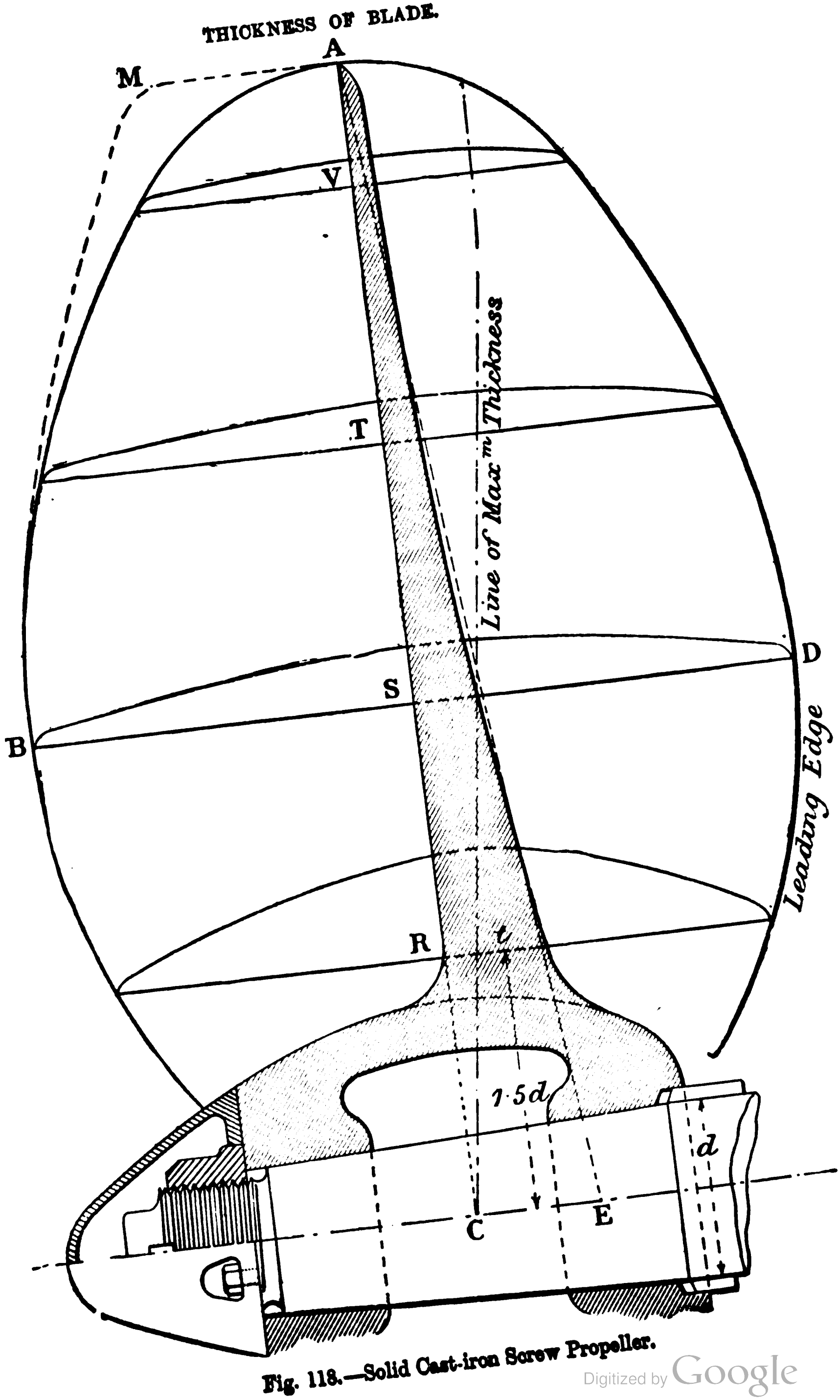


Fig. 118.—Solid Cast-iron Screw Propeller.

Fig. 118 is a typical one of a solid cast-iron propeller. It is also typical of the form of the blade of a loose-bladed propeller, and the proportions are such as would be found generally with those having three blades. The Admiralty, as well as many private engineers, prefer to have the blades rounded at the top as shown, but blades in the past, as well as many of them in the present, are shaped as shown in dotted lines at M.

The surface of propellers designed approximately to fig. 118 may be found by the following rule:—

$$\text{Aggregate surface} = D \times B \times n \div F.$$

D being the diameter in feet, B the maximum breadth in feet, n the number of blades, and F for rounded tips, 3, and for square ended, 2·85.

The blade sections are obtained by taking a point E, so that

$$OE = \sqrt{\frac{d^3}{n \times b}}\, K^1,$$

d being the diameter of the screw shaft at the inner end or that of the tunnel shafting, n the number of blades, b their maximum breadth in inches. The value of K^1 is, for cast iron, 6·3; for ordinary gun-metal or bronze, 3·2; for cast steel, 2·5; and for bronzes of superior strength, 2·35.

The transverse sections of the blade should be somewhat like the water-lines of a ship, with a fore-body shorter than the after-body; the line of maximum thickness is shown on the figure, and the sections should taper in thickness from that line on either side. Some engineers do so by straight lines, but a slight amount of curvature is better.

Taking T as the thickness at $\frac{1}{9}$th of the diameter, or 1·5 d from the centre, the thickness at $\frac{2}{9}$ths should be 0·65 T, at $\frac{3}{9}$ths = 0·4 T, and at $\frac{5}{9}$ths = 0·23 T.

Propeller Boss.—When for a loose-bladed propeller, this is usually spherical in general form, with flats or recesses for the blades. The diameter of the sphere is from $\frac{1}{4}$ the diameter of the screw for small propellers, to $\frac{1}{6}$ the diameter for large ones, and may be taken at

$$= 0{\cdot}9\sqrt{\text{diameter screw}}.$$

The length of the boss depends on the size of the base of the blades, and is generally about 0·85 the diameter of sphere for a two-bladed screw, and 0·75 when four-bladed.

The bosses of some very large screws have been made oval in fore and aft section, and the base of the blades made oval to fit them. This form is not so good for many reasons as the spherical, and was adopted more on account of the boss being a forging than from any other cause.

In H.M. Navy, as well as in the navies of most countries, bronze of some kind is the material employed for both boss and blades. The Admiralty specify that it must be composed of 87·65 best selected copper, 8·32 tin, and 4·03 best Silesian spelter. This mixture, when carefully made, should have an ultimate tensile strength of 16 tons per square inch, and be very tough. Unless very carefully melted, however, test pieces from large castings seldom exceed 14 tons.

The Admiralty have also phosphor-bronze and manganese-bronze propellers, but seem not to prefer them to ordinary bronze, which possibly may be more suitable to the service, and less liable to deterioration, &c. Bronze is a necessity with some naval ships, on account of the copper on the bottoms and the brass elsewhere in the submerged parts affecting cast iron or steel; it is also used because its strength is superior to that of cast iron, and because steel is viewed with some considerable amount of distrust by the authorities, which has not been lessened by the action of some of the larger steamship companies, who are now generally adopting bronzes in lieu of steel.

In the mercantile marine, the boss is generally of cast iron, of as strong a mixture as can be made, or of cast steel. In large steamers and all ships which make long voyages to distant parts the bosses should be of cast steel, whether the blades be of iron, steel, or bronze.

The studs for securing the blades to the boss are of bronze for bronze blades, and sometimes for iron and steel blades, but for the latter best iron or steel is preferable, the nuts being, however, of brass with closed ends to protect the studs.

The bosses of solid cast propellers are oval in fore and aft section as a rule, and are from ⅙ to ⅕ the diameter of the screw in diameter, and about 2½ to 2¾ times the diameter of the shaft in length.

Screw Propeller Blades.—The screws of ships of the mercantile marine are generally of cast iron, and up to moderate sizes are almost invariably cast solid, although latterly there is a decided tendency to adopt those with loose blades. A solid screw is more efficient than one whose blades are bolted on, and is only about half the cost of the latter. The cause of inefficiency arises from the resistance of the projecting nuts, and often this is added to by the clumsy flanges of the blades, which project beyond the boss itself. A well-designed and carefully made screw should have the base of the blade conforming to the general outline of the boss, and the nuts or bolt heads recessed into the blade base, and covered in with a metal case or cement flush with the surface (fig. 118*a*). The screws in H.M. Navy are generally made in this way, and are then quite as efficient as a solid screw.

If one of the blades of a solid screw is broken off, the whole screw is practically ruined,* and the expense of a new one thereby entailed; also, what is of more serious consequence, the ship must go into a dry dock, or on a slip or "hard" to have it removed, which operation is long and tedious, as the boss must be forced off, the tunnel-shaft taken down, and the screw-shaft withdrawn before the new one can be fitted. On the other hand, if the screw has its blades bolted on, and one is damaged, it can be removed by a diver, and a new one fitted in its place at a very small expense, and in a very short time; and even when an experienced diver cannot be obtained, the ship can be tipped by discharging cargo from aft, &c.

The forms of propeller blades are so numerous and varied as to be beyond description here. The well-known blade (fig. 119) introduced by Mr. Griffiths, and now bearing his name, was the one most generally adopted by engineers, and generally gave satisfaction. Only a few carry out the plan he usually so strongly recommended, of bending the blade slightly forward, perhaps principally because it then came too near to the stern-post. Some have tried blades bent in the reverse way, and satisfied themselves that improved performance is obtained. The data published by inventors of screws are generally very misleading, as the failures are never mentioned, and seldom is it observable that an old screw has been replaced by a patent one of the same diameter, pitch, and area. There is little doubt that the

* A clever founder can burn on new ends to blades both with iron and bronze.

better results with the "improved screw" are due rather to better proportions than to the particular *shape* of blade. The number of patents relating solely to the *form* of blade is endless, and some special ones, introducing additions "to prevent loss from the centrifugal action," reappear periodically as novelties. This latter is a great bugbear with many engineers; for after all, the loss from this cause is in a well-formed screw very slight, so that it always happens that means adopted to prevent it are themselves causes of a greater loss from frictional resistance.

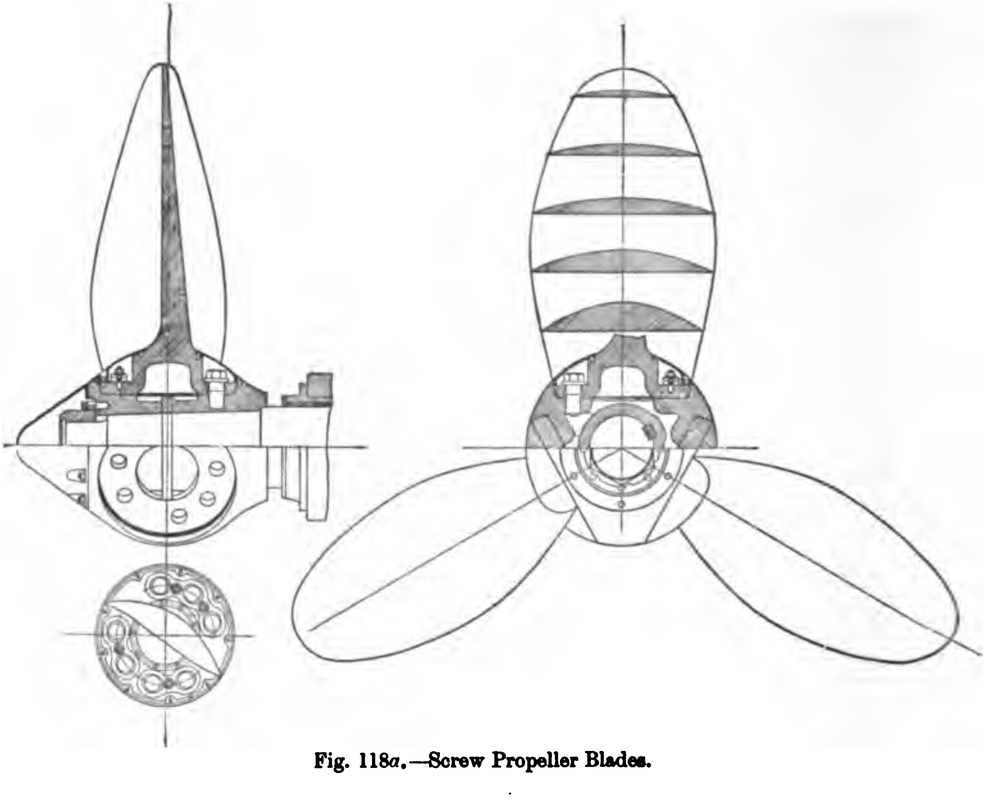


Fig. 118*a*.—Screw Propeller Blades.

As a rule, the greatest breadth of blade should not be beyond one-third of the diameter of the disc from the centre, and should be approximately as given by the following rule:—

$$\text{Maximum breadth of blade in inches} = K\sqrt[3]{\frac{\text{I.H.P.}}{\text{revolutions}}}.$$

For a four-bladed screw, K = 14; for three-bladed, K = 17; and for two-bladed, K = 22.

The breadth of blade at the tip should be from $\frac{1}{5}$ to $\frac{2}{5}$ the maximum.

Propellers are often made with the blades of finer pitch near the boss than at the tip, partly that the angle shall not be so coarse that this part of the blade only churns the water, and partly that the hold on the boss due to the increased breadth of the blade may be greater. A decrease of pitch of 10 per cent. has given good results, and when the propeller is of small diameter, with a very coarse pitch, as much as 15 per cent. decrease may be adopted; as a rule, however, a true helix gives best propelling results.

Material for Screw Blades.—Cast iron is, of course, the cheapest material for this purpose, and also possesses another advantage not so much appreciated until experience teaches—viz., that when struck violently against an obstacle like a jetty or wreckage, it breaks clean off, without, as a rule, damaging the shaft; on the other hand, steel or bronze blades, which are strong enough and tough enough to resist fracture, are sometimes bent so as to prevent the screw from turning, and often cause the shaft to be seriously bent, or even broken. Cast iron when used should be of the very best description, twice cast and cooled slowly; hematite, and even steel, is often added to strengthen the mixture; the former is now generally used by moulders for this purpose.

Steel blades can now be bought at about two and a-half times the cost of cast-iron ones; they are very much stronger, even when the section is considerably reduced, and consequently are more reliable, especially for engines of very large power, which must work at high speeds in rough weather. The efficiency of the propeller is also much increased in consequence of the reduction in thickness; hence when the best possible results are necessary blades of the strong bronzes should be fitted with their surfaces ground or filed smooth.

The chief objection to steel, especially to cast steel, is its liability to rapid corrosion on the backs of the blades. Cast iron corrodes into pits, but steel goes much more rapidly, and in some instances the tips are "honeycombed" in a few weeks.

Paint provides little or no protection, and even nickel plating, which has been resorted to by some, has not proved very efficacious. Muntz metal sheathing has been tried, but with not very satisfactory results.

Phosphor-bronze, Parson's manganese-bronze, Stone's bronze, Delta metal, &c., are taking the place of steel in many quarters, notwithstanding that the cost is about twelve times that of cast iron. Blades made of these materials can be cast very thin—thinner, in some cases, than if made of steel—because there is no loss of strength by corrosion; in fact, the very high speed of certain ships is attributed largely to the high efficiency of the propeller from the thinness of the blades.

Bronze propellers are objected to on the ground that injury is often done to the iron work near them by galvanic action; but the Admiralty does not hesitate to continue the practice of fitting them, nor do the large steamship companies. Corrosion does sometimes take place in such a way as to cause the source of the evil to be attributed to bronze screws; but the fitting of some slabs of zinc in way of the propeller prevents it, and removes all fear of ill consequences.

Weight of Screw Propellers may be calculated with a close approximation to truth by the following rule:—

$$\text{Weight in cwt.} = \frac{\text{Surface} \times \text{thickness}}{K}.$$

The surface is taken in square feet, the thickness in inches at the root of the blade.

K is for solid cast-iron screws, 4·5
" " bronze " 3·8
" built C. iron " 3·0
" " C. steel " 2·8
" " bronze " 2·5

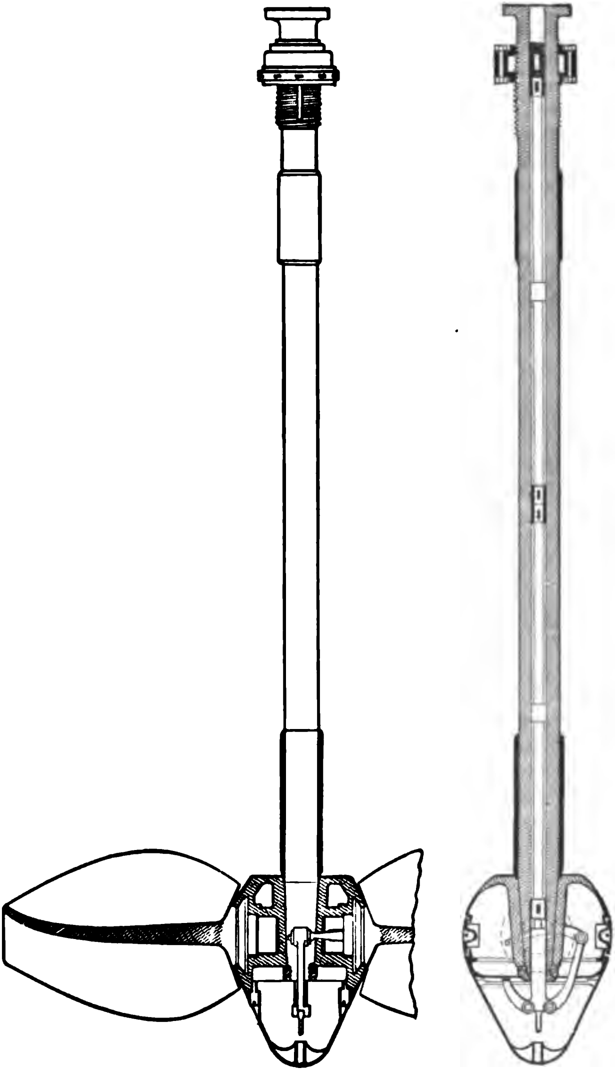


Fig 119.—Bevis Patent Feathering Screw.

Feathering Screws.—Yachts and ships which are required to sail as well as steam cannot well do the former when the screw is stopped, unless some means be adopted of feathering the blades, so that they are nearly in a fore and aft plane, or else by withdrawing the propeller altogether from the water. The late Bennett Woodcroft patented, in 1844, a plan for feathering the blades, which in a modified form was fitted by Messrs. Maudslay, Sons, & Field to several ships. The blades, of which there are two, have shanks fitting into the boss, to which short levers are secured inside the boss; these levers are connected by links to a sliding collar outside the boss, which is carried round with the shaft, but is capable of being moved "fore and aft" on it by means of a pair of bell-crank levers actuated by a screw from on deck. When it was desired to sail, the blades were moved round into the fore and aft position by sliding the collar from the boss.

Bevis' Patent Feathering Screw.—Many patents were taken out for methods of effecting a similar movement of the blades, without the objectionable feature of external bell cranks, and to the late Mr. Bevis is due the perfecting of this idea. In 1858 Gregory and Craymer patented a feathering screw of which "the screw propeller shaft is made hollow, and a second shaft goes through it, carrying worm threads, which act on the propeller blades, so as to feather them to the angle desirable." In 1866 H. B. Young patented a somewhat similar idea as to the hollow shaft, but says "levers may be attached to the shanks." Fig. 119 shows the Bevis plan, which needs no description, and answers its purpose admirably.

Lifting Screws.—The screws of warships were formerly nearly always made and fitted, so that when desired they could be raised to the level of the deck for examination and repair when necessary, and to prevent obstruction when sailing. This plan is a very costly one, and not so efficient as the feathering blade, inasmuch as the ship steers badly owing to the gap in the deadwood, but it admits of examination and repair, which is of the utmost importance in a warship. A hole, however, is also necessary through the stern to admit of the screw coming on deck, which very much weakens what is already a somewhat weak part of the hull.

The lifting screw has a short piece of shaft cast with the boss or fitted to it in the usual way; the forward end of this shaft is provided with a driving piece, which is so formed as to fit into a slot across the cheese coupling keyed to the outer end of the stern shaft. The propeller is carried in a frame, called the *banjo frame*, which is arranged to slide up and down in grooved metal guides secured to the stern and rudder posts, and supported, when in working position, by two strong brackets or chairs also secured to these posts, and held down by two strong wooden sampson posts, whose upper ends are fitted with jam screws to the metal guides; such an arrangement is made wholly of gun-metal.

CHAPTER XVI.

SEA-COCKS AND VALVES AND AUXILIARY MACHINERY.

The importance of having efficient connections to the skin of the ship for inlet and outlet purpose has long been recognised, since neglect in the design or working of these fittings has sometimes caused the loss of a ship. Next to absolute safety, simplicity should be most aimed at in all parts of the machinery of a ship; but in no place, perhaps, is there greater need of careful consideration than in the arranging and designing of the sea and bilge fittings, so that neither a careless nor an ignorant engineer can endanger the safety of the ship.

Lloyd's Committee have issued special rules on this subject, and the Board of Trade are no less vigilant in endeavouring to eliminate every source of danger.

It is needless here to give any illustration of how easily a ship may be flooded by an ill-considered arrangement of sea-cocks in ignorant hands, nor to give examples of ships that have foundered, for every engineer has heard of them. But even good arrangements may cause mischief, if, from want of simplicity, they are ill understood.

In the first place, the cocks and valves themselves should be of simple construction, and carefully marked, so as to be easily seen whether they are open or shut. The parts exposed should be either strongly made or carefully guarded, so that they are not liable to injury or derangement; and, when possible, so designed that they can be packed and examined without having to dock the ship. It is also necessary that the very large openings should have a second valve, as a safeguard in case of accident to the first one.

Sea-cocks and valves should be so placed that they are accessible at all times; and, as far as possible, within sight from the working platforms.

Kingston Valve.—For all large inlets the *Kingston valve* is preferable, as it acts as a non-return valve in case of the spindle breaking, and can then always be worked by simply forcing it outwards, either with the spindle itself, or by a rod substituted for it.

The Kingston valve is usually made in the form of a frustrum of a cone, with a taper of about 8 in 12; the length of the seat is generally about $1 \text{ inch} + \frac{\text{diameter}}{12}$. The object of this form of seat is to allow the valve to close itself tight in place, in case of the spindle breaking, by the pressure of the water. The Admiralty require that these valves shall be made in one with their spindles, of best gun-metal, and the spindle *tested* to a strain of half a ton for each square inch of section of valve. For example, a valve 4 inches diameter, having an area of 12·56 square inches, should have a spindle sufficiently large to be tested to $6\frac{1}{4}$ tons.

Hence, if the proof strain of good gun-metal be taken at 6 tons (which is quite high enough) per square inch, then,

$$\left.\begin{array}{l}\text{Diameter of Kingston valve spindle at bottom}\\ \quad\text{of thread,}\quad . \quad . \quad . \quad . \quad .\end{array}\right\} = \frac{\text{diameter of valve}}{3{\cdot}46};$$

so that a 4-inch valve should have a spindle 1·15 inch diameter at its smallest section.

But the Admiralty do not require any valve to be tested above 12 tons; so that for mere test purposes, no spindle need have more than 2 square inches in section, or be more than 1⅝ inch diameter at its smallest part; but since for very large valves a spindle of this size would not be stiff enough, the following rule for all valves above 5½ inches diameter holds good:—

$$\text{Diameter of spindle} = 1\tfrac{5}{8}\text{ inch} + \frac{\text{diameter in inches} - 5\tfrac{1}{2}\text{ inches.}}{16}$$

Kingston valve-boxes and tubes are generally made of gun-metal; but this is not a necessity, except where hot water or steam is blown through them, when cast iron would be dangerous. If the body is of cast iron, of course the valve seat should be of brass, and the working parts bushed with brass.

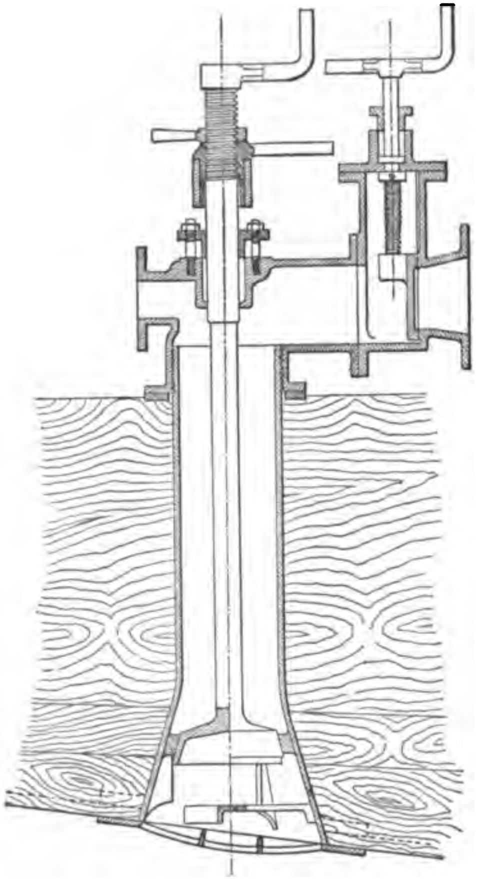
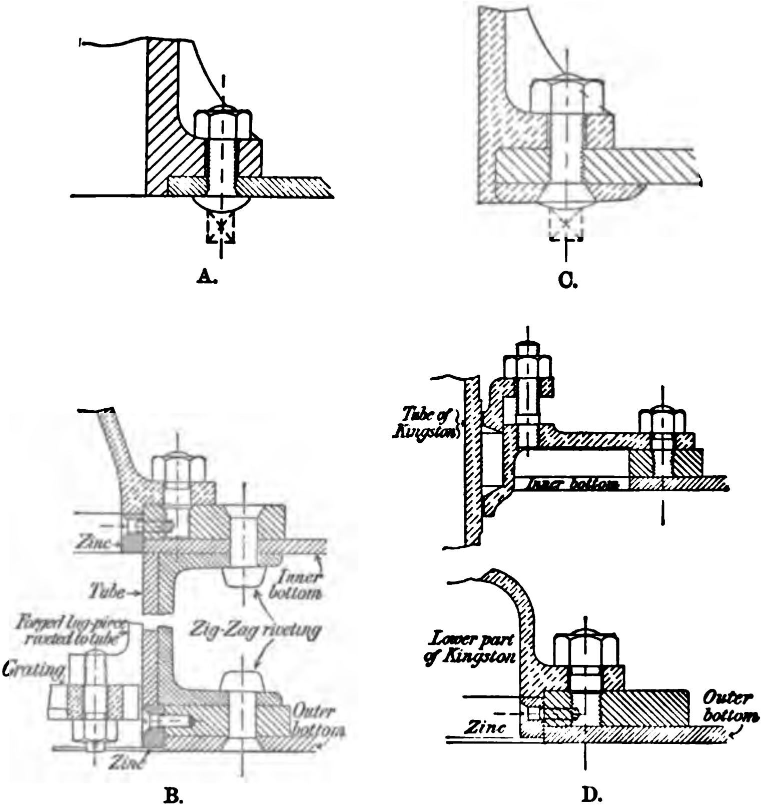


Fig. 120.—Kingston Valve.

Fig. 120 shows a good arrangement of Kingston valve for large sizes. It has a lifting nut secured in a bridge, as well as a handle on the spindle end; the former is used to start the valve or jam it in its seat, and the latter merely to open or shut it; the lock nut with handle is to secure it in any required position. In this figure the head is shown screwed on to the tube, and is such as was necessary in wooden or composite ships; but when fitted to the skin of an iron ship, a flange is formed at the bottom of the conical part, as shown by dotted lines and in Fig. 121 (A, B, C, D). A much simpler and less expensive plan is to form the valve like an ordinary stop-valve with four wings, and the spindle inverted—that is, on the same side as the wings; the bottom part of the box is then only slightly conical, and much shorter —in fact, only sufficiently long to allow of the valve lifting a distance equal to one quarter of its diameter, and in this position leaving space between it and the grating for the free flow of water.

All inlet valves should be fitted with a brass grating, whose meshes should not exceed half an inch in breadth, the total area through them being at least 20 per cent. larger than the net area of the tube. In the Navy, Kingston valves are fitted to all inlets and blow-off pipes, and are always supplemented with a cock or valve attached to them.



A. C.

B. D.

Fig. 121.—Details of Inlet Valves.

Valves are fast taking the place of cocks for all general purposes, so that whenever convenient a valve may be fitted in lieu of a cock to even the smallest Kingston, but to large Kingstons a cock could not be fitted, and so it is usual to find an ordinary sluice valve. The spindles of these supplementary valves should always be within easy reach of the platforms, and in case of the very large ones, when possible, they should be carried so high as to admit of the valve being worked when the engine-room is flooded.

In the merchant service, whether valves or cocks are fitted to the skin of the ship direct, the same precautions should be taken as enumerated above, and extra care taken to protect them from injury, as they are very liable to damage from the flooring plates washing about when a considerable quantity of water is in the bilge and the ship rolling heavily.

For the larger inlets of a merchant ship an ordinary stop-valve, opening inwards, is usually fitted, the box being of cast iron. A good plan, so as to avoid a number of openings in the ship's skin, is to fit a single box with one stop-valve in it, and that by preference an inverted one, and to this box fix the various cocks or valves necessary for the different requirements. For ships destined for a cold climate, a small cock can be fitted near the bottom of the box to admit steam to thaw any ice that may be blocking the orifice.

Discharge Valves should be fitted to all outlets through the ship's side, and they consist, as a rule, of simple non-return valves having spindles passing through the covers, so that the valves may be lifted or pressed down as required. Here again a number of holes through the skin of the ship may be avoided by connecting the smaller valves to the box of a large one above the valve itself. In many cases this method is a very good one, having advantages beyond that already named—viz., that the pipes can be shorter, and the valves standing clear of the frames of the ship are easy of access.

The Board of Trade * require that all inlet and discharge valves shall be connected directly to the skin of the ship, and have no pipe or joint intervening (this rule, however, does not apply to the case of the smaller valves when attached to the box of a large one), and also that all discharge valves shall be placed above the load water-line.

The Admiralty, on the other hand, for obvious reasons, require all the discharge orifices to be below the water-line; and while in the merchant service the discharge valve-boxes are nearly always of cast iron, in the Navy they are invariably of brass.

It is not unusual to fit the smaller discharge valves as simple non-return valves having no external spindle whatever; but this has the disadvantage that one is never certain if the valve is shut, and if not shut it is beyond control.

It is sometimes found convenient to fit, for large discharges, a *straight-through* valve in lieu of the ordinary mushroom-valve. This straight-through valve is an ordinary flap valve resting on a vertical or nearly vertical seat, and lifted by means of a horizontal spindle, which passes through a stuffing in the side of the valve-box, having inside a slotted lever connected by a pin to a pair of lugs on the back of the valve, and, outside, a hand lever for controlling the valve.

Bilge Valves.—As has been already said, too much care cannot be devoted to avoid all risk of flooding the ship by carelessness. An easy method of overcoming the difficulty in the case of pipes not greater than

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* The Board of Trade regulations referring to sea connections are as follows :—

137. All inlets or outlets in the bottom or side of a vessel near to, at, or below the deep-load water line other than the outlets of water-closet, soil, scupper, lavatory and urinal pipes, should have cocks or valves fitted between the pipes and the ship's side or bottom ; such cocks or valves should be attached to the skin of the ship, and be so arranged that they can be easily and expeditiously opened or closed at any time; and the cocks, valves, and the whole length of the pipes should be accessible at all times.

Cocks or valves standing exceptional distances from the ship's plating, that is where the necks are longer than is necessary for making the joint, should not be passed without the sanction of the Board of Trade, and one condition of their being passed is that they should be made of gun-metal and well bracketed.

4 inches diameter, is by means of the switch or three-way cock—that is, a cock having two side branches and a hollow plug with one orifice in the side and the other in the bottom, so that all water must pass by way of the bottom and one side only at one time. Suppose that an auxiliary engine is required to draw water from the bilge as well as from the sea, a three-way cock should be fitted so that its bottom is connected to the pump suction, one side branch connected to the sea inlet, and the other side branch to the bilge piping; it will then be easily seen that the two branches cannot be connected by the plug, and consequently, however careless the engineer may be, water cannot pass from the sea to the bilges. When, owing to the largeness of the piping, a cock cannot be fitted, non-return valves should be connected to the tail pipes leading to the bilge.

Communication Boxes.—All the bilge suctions, and all ballast-tank suctions, should be led to a *communication box* placed in some convenient position in the engine-room, above the level of the flooring, so that the requisite changes of service of pumping may be easily and quickly effected.

The communication box (called sometimes a "directing" box) consists of a rectangular chest having as many valves in it as there are pumping stations or parts from which water is to be drawn; under each valve is a separate tail, to which the suction pipes are attached; the valves themselves are of the ordinary mushroom type, non-return, and arranged so that each may be shut close by screwing down a spindle on it. The box cover is so fitted by hinge bolts as to be easily and quickly opened for examination, and the joint made again.

To the upper part of the box the suction pipes of the pumps are fitted, so that the auxiliary pumping engines and engine bilge-pumps may draw from the same set of pipes. Both bilge-pumps should not be connected direct to one box, as it not unfrequently happens that it is required to pump water separately from two compartments at the same time, or that one of the bilge-pumps may be required when no sanitary pump is fitted to deliver on deck. The lid of the communication box should have inscribed legibly on it the lead of each valve.

Bilge Suction Piping is sometimes of cast iron and sometimes of lead, but the tails leading directly into the bilge should be of iron; cast iron is now often used instead of lead as less liable to damage, but it cannot be so easily cut in case of necessity. The Admiralty require all tail pipes to be of galvanised wrought iron; and all copper piping laid in the bilges to be covered with waterproof canvas varnished over, and otherwise insulated to avoid deleterious effects on the ironwork from galvanic action. All suction pipes to the bilge should fit into rose-boxes of iron galvanised, and having an aggregate area through the holes at least twice that of the pipe. The cover of the box should be hinged, or so fitted that it can be opened and shut again very quickly. The boxes should also be so placed as to be easy of access. Bilge pump suction pipes should also be fitted with mud traps, consisting simply of a cast-iron box with a strainer and lid, in which the slight cessation of flow allows the deposit of the heavy dirt carried on so far with the water. All copper or brass-work subject to the action of bilge-water, should have the connections made with *bronze* bolts and nuts. Iron quickly corrodes unless well protected; and Muntz metal has been found to decay in a very peculiar way, becoming so soft as to cut like plumbago. Naval brass, which is Muntz metal improved with a little tin, is suitable for this purpose. Every care should be taken to protect the metallic surfaces from the corrosive action of bilge-water, a good coat of Portland cement wash answering even better than paint or varnish.

When it is required to pass pipes, especially copper ones, through a watertight bulkhead, a good plan to adopt is to fit a short length of cast brass, having a flange at each end to connect to the pipes on either side, and a collar in the middle, about 4 inches larger in diameter than the flanges, to secure it to the bulkhead.

If pipes are likely to expand considerably, or to be in such a position that the working of the ship in a seaway would affect them, it is better to pass them through stuffing-boxes in the bulkhead. Very small pipes can be connected by union nuts to the bulkhead joint above described, thereby decreasing the size of the hole to be cut and the collar for closing it.

Auxiliary Pumps.—In the merchant service, in small ships, one donkey engine is often found sufficient for all purposes, and consequently its pipes should be so arranged that it can draw water from the sea, the bilges, and the hotwell; and deliver to the boilers, overboard, and to the deck. In many cases, too, this donkey engine is fitted to pump into the condenser, that it may be charged with cold water before starting the engines. In the mercantile marine, as in the Navy, it is now very general to do without feed-pumps worked by the main engines, and to have instead one or more independent pumps of the Weir type devoted solely to feed the boilers, and another or auxiliary pump of the same or of the "Duplex" type as a reserve for the same service, and delivering sea or reserve water on deck. In addition to these is a fourth auxiliary pump of either type, arranged to draw from the sea, bilges, or tanks, and deliver on deck or overboard. In large steamers having capacious water ballast tanks, a fifth pump for the purpose of emptying these tanks is very generally supplied; it is also so arranged as to do duty as a bilge pump, and delivers overboard only. In all ships it is a great convenience to have an *ejector* from 1½ to 2½ inches to keep the bilges quite clear of water with the least trouble. In the Navy there is always a multiplicity of donkey engines; four for supplying the boilers only; and two others, called usually the fire engines, pump from the sea to the deck; and two more arranged to draw from the bilges overboard. Latterly a pump has been fitted in each engine room of naval

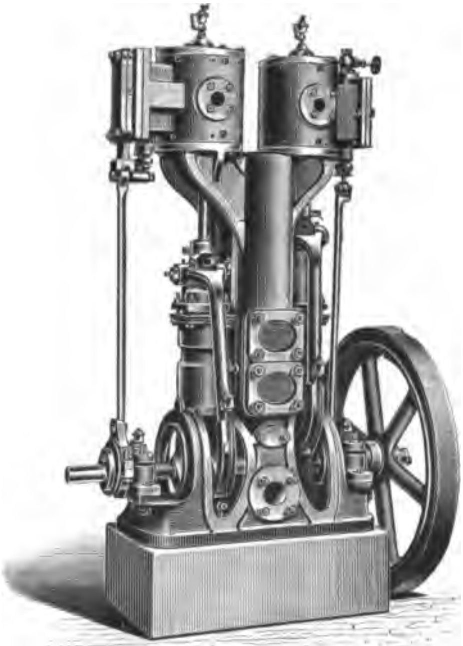

Fig. 122.—Pearn's Double Ram Feed Pump.

ships to draw water from the hot-well and deliver to the feed tanks. If these pumps were arranged to drain the condensers of water and deliver to the feed tanks, a better end would be attained, as the air-pumps would be relieved of the liability to strain from being glutted with water; they would also prevent accumulations of water.

In addition to the donkey engines there is usually a hand pump, which, in the Navy and large merchant ships, is so arranged that it can be worked from the main engines, and has pipes, &c., fitted, that it may do the same duties as the donkey engines; and in the Navy, in addition to this, the boilers can be emptied by it, instead of being "blown down."

The Board of Trade require that one of the bilge-pumps shall be so fitted that it can pump water on deck in case of fire.

The engine-room pumps should have the necessary piping, valves, &c., that water may be drawn from each hold or compartment separately, from each side of the engine and boiler rooms, and from any other part of the ship liable to leakage or lodgment of water.

In addition to the above-mentioned means of freeing the ship from water, it is usual to have a bilge suction, so that the circulating pump may be utilised. The chief danger to be apprehended when the circulating pump draws from the bilge is, that in cases of flooding of the engine compartment, the small coal is washed out of the bunkers, and soon chokes the condenser tubes as well as the pumps themselves, unless the

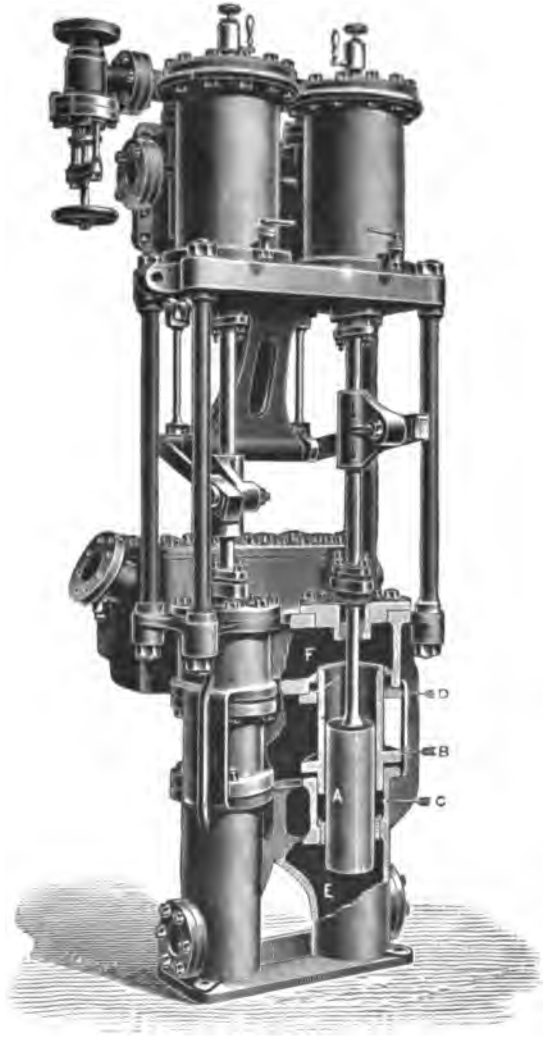

Fig. 123.—Pearn's Patent External Packed Pumps (Duplex).

rose boxes have sufficiently small holes to prevent it; and when the rose-box holes are so small as to prevent small coal passing through them, they soon choke, and render the pumps useless for the time. With a centrifugal circulating pump there is no danger of the pump choking or getting out of order, and by having a direct discharge overboard, the condenser cannot get choked.

Donkey Pumps are now usually of the direct type—that is, the steam piston is connected direct to the water piston, and works without the intervention of a crank-shaft and flywheel. The steam distributing valve of these pumps is generally made to reciprocate by steam pressure acting on its ends alternately, and is known as a *shuttle valve;* the steam to its ends is regulated by a small auxiliary valve operated by the pump-rod or the steam piston itself. The mechanical efficiency of such pumps, owing chiefly to the absence of gear, is naturally very high, and in great measure compensates for the low steam efficiency due to the cut-off being at or near the end of the stroke. In fact, the combined efficiency of some of these pumps having very long strokes is remarkable. Their maker, however, admits that a saving of 20 per cent. is effected by compounding with tandem cylinders. The earliest of these direct pumps had only a single steam cylinder and one double-acting pump; the small valves influencing the shuttle were moved by contact with the cylinder ends, and cushioning was effected by the piston closing the exhaust port before the end of the stroke, steam being admitted by a separate port, which was closed by the shuttle valve when exhausting. The pumps had and have the merit of always making the full stroke and being cheap. Mr. Weir introduced his pump about 25 years ago, placed vertically, instead of horizontally, as the others had been; the shuttle valve of his steam cylinder had, however, a horizontal movement, its influencing valve having a vertical movement imparted to it by a lever, connected to the piston-rod; since then he has improved and perfected it, so that it has the confidence and good opinion of all sea-going engineers. Other makers have followed Mr. Weir in placing on the market somewhat similar pumps, differing only in details. The now well-known Worthington pump of American origin consists of two sets of cylinders and pumps direct, driven and placed side by side, so that the steam valve of one can be operated by a lever moved by the other, the valve in this case being of the ordinary kind; when in good order and properly set nothing can be more admirable than the way in which this pump works. If, however, the adjustments have been carelessly made, or the wear of packings, &c., has been uneven, the pumps do not always make the full strokes. Fig. 123 shows a vertical pump of the Worthington type having means for adjusting the packing of the pump plungers while the pump is working, thereby maintaining its efficiency without putting it out of service; this ingenious method is the invention of Mr. Frank Pearn, whose double pumps (fig. 122) with crank and flywheel are perhaps the best of that type in the market, and whose single pumps of wall type are such favourites for feeding small boilers. Auxiliary pumps should have bronze water ends and fittings, the plungers or pistons packed with ebonite or woodite, the rod of rolled bronze of best quality, for standing shock, and the valves small and numerous so as to give the maximum amount of perimetrical opening with small lift; they also must be made of a bronze that will withstand continuous shock, and, by preference, machined from rolled or forged rod.

Injectors for feeding marine boilers have not so far been looked on with favour; why, is not apparent, for they return to the boiler all the steam they take, and consequently their efficiency is high, and they certainly are convenient and easy to use for supplying sea-water or cold water from reserve tanks.

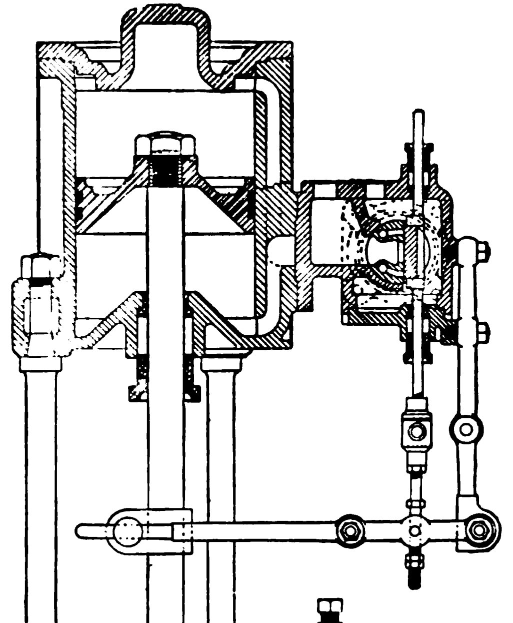

Fig. 124.

Water Service.—An arrangement of water service is made in every ship, so that water can be applied to all bearings and slides, and also that a hose may be used in case of fire or any other emergency. This generally consists of a main pipe from a sea-cock leading into a convenient position, and having branches with a stand-pipe and brackets at each main bearing, crank-pin, and group of eccentrics. The water for the tunnel-bearings and thrust-bearing is usually taken from the inside of the stern-tube, thereby serving the additional purpose of circulation in the tube.

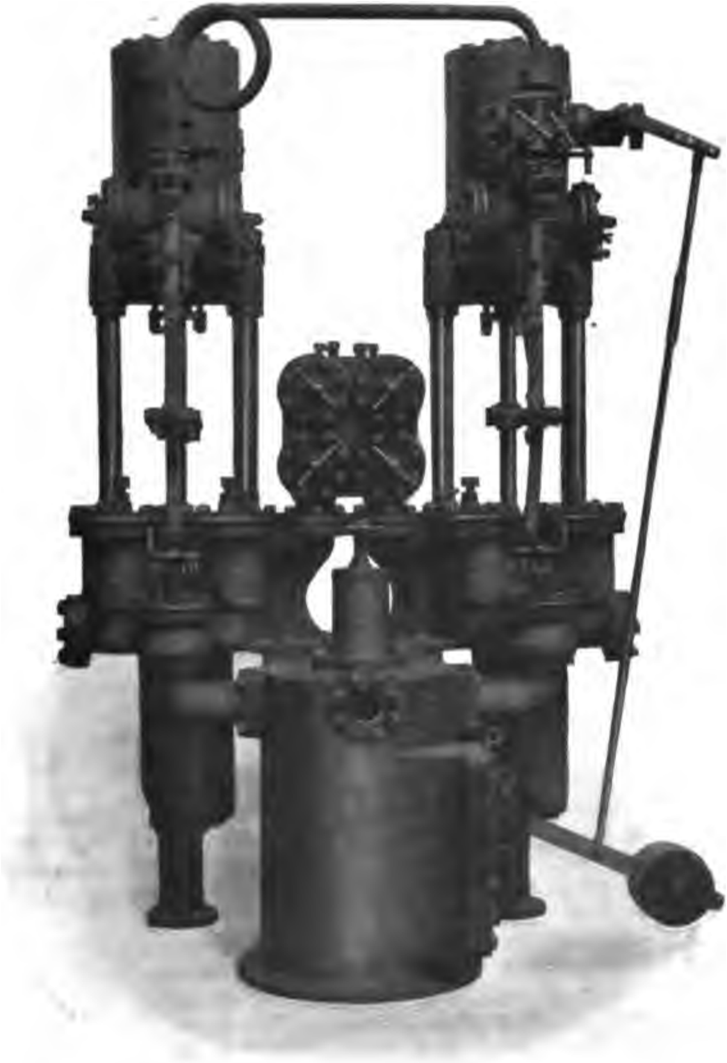

Fig. 125.—Weir's D. A. Feed Pumps, with Float Tank and Automatic Gear.

Expansion Joints.—Great care should be taken that all pipes conveying hot water or steam, and thereby liable to great change of temperature, should have provision made for the consequent expansion. Expansion joints of all kinds are, however, objectionable on one or two grounds, and therefore should be avoided unless absolutely necessary. If small pipes have bends between the rigid connections, the expansion is allowed for by the increase in their curvature; but in all large pipes, and wherever the bends are but very slight, expansion joints should be fitted, for very serious accidents have occurred in consequence of their absence. When the length of piping to undergo expansion is not great, the ordinary bellows joint is preferable, as being free from any liability to leak; but when the expansion is very great an ordinary stuffing-box and gland, or Fawcett joint, must be resorted to. This latter should be avoided for exhaust pipes to a condenser, as they are very liable in this case to leakage which is not easy to detect. A thick india-rubber jointing ring will generally serve this purpose with eduction pipes; in the case of large ones this may be 2 inches thick.

Again, in ships of light construction, which undergo a considerable amount of racking in a sea-way, no length of pipes should be without some provision for expansion and contraction, and all extra rigidity should be avoided, that they meet the changes in ship. Great care should be taken to provide means for resisting the thrust of pipes which are subject to internal pressure or heat, otherwise the joints are severely strained, and the connection of pipe to flange and of valve-box to boiler in danger of destruction. This is the more necessary when bends are expected to do duty for expansion-boxes, &c.

Safety Collars.—It has sometimes happened that pipes have blown out of a Fawcett joint, owing to there being a bend near it on which the steam pressure acted; in all cases of this kind a collar should be brazed to the pipe, not less than 18 inches from the bend, through which bolts pass connecting it to the flange of the stuffing-box on the other pipe, that it may not draw out.

Flanges.—Few things look worse in an engine-room than heavy broad flanges to the copper pipes, especially as they are quite unnecessary. The breadth of the flange need not exceed three times the diameter of the bolts through it, and its thickness should equal the diameter of the bolts. The pitch of the bolts should be from four to six times their diameter, depending on the steam pressure.

The Admiralty require all copper pipes to be in accordance with a graduated scale.

Table xxiii. (p. 349) has been drawn up as a general rule indicating the best thickness of pipes for various pressures and purposes, and may be taken as giving thoroughly reliable strengths for all ordinary marine work.

In the merchant service the main steam pipes, when of copper, are usually from No. 1 to 4 B.W.G. thick, depending on their diameter and the pressure to which they are exposed.

BOARD OF TRADE RULES FOR PIPES.

117. The working pressure of well made copper pipes when the joints are brazed is found by the following formula :—

$$\frac{6000 \times \left(T - \frac{1}{16}\right)}{D} = \text{working pressure};$$

T = thickness in inches;

D = inside diameter in inches.

When the pipes are solid drawn and not over 10 inches diameter, substitute in the foregoing formula $\frac{1}{32}$ for $\frac{1}{16}$.

118. The internal pressure on wrought-iron pipes made of good material, which are lap-welded and are a sound job, may be determined by the following formula, provided that the thickness is not less than ¼-inch:—

T = thickness in inches;
D = diameter inside in inches;

$$\frac{6000 \times T}{D} = \text{working pressure.}$$

119. Feed pipes should be made sufficient for a pressure 20 per cent. in excess of the boiler pressure.

TABLE XXIII.—Thickness of Copper Pipes (L.S.G.).

Diameter of Pipe in Inches.	Steam Pipes.						Auxiliary Exhaust Pipes.	Waste Steam Pipes.	Main Water Pipes.	Bilge Suction and Disch. Feed Suction and Fire Service.
	Boiler Pressures in Lbs.									
	200	180	155	125	85	50				
22	...	...	...	...	0	7	...	...	4	...
21	...	...	...	...	1	7	...	...	4	...
20	...	...	...	...	1	7	...	...	5	...
19	...	...	...	...	2	8	...	...	5	...
18	...	...	...	4/0	2	8	...	13	5	...
17	...	...	6/0	3/0	3	8	...	13	6	...
16	...	7/0	5/0	2/0	3	8	...	14	6	...
15	...	6/0	4/0	0	4	8	...	14	7	...
14	7/0	5/0	3/0	0	4	9	11	14	7	...
13	6/0	4/0	2/0	1	5	9	11	14	7	...
12	5/0	3/0	0	2	5	9	11	14	8	...
11	4/0	2/0	1	3	6	9	11	15	8	...
10	3/0	0	1	3	6	10	11	15	9	...
9	2/0	1	2	4	7	10	11	15	9	...
8	1	2	3	5	8	10	12	15	9	...
7	2	3	4	6	8	11	12	15	10	9
6	3	4	5	7	9	11	12	15	10	10
5	5	6	6	8	9	11	12	15	11	10
4	6	7	7	9	10	12	12	15	11	11
3	8	9	9	10	11	12	12	15	12	11
2	10	11	11	11	12	12	12	15	12	12
1	13	13	13	13	13	13	13	15	13	13

Diameter of Pipe in Inches.	Main Eduction and Air-Pump Suction.
35	3
33	3
32	4
29	4
28	5
25	5
24	6
21	6
20	7
17	7
16	8
15	8
14	9
13	9
12	10
11	10
10	11
9	11
8	12
7	12
6	13
5	13

Blow-off and Scum Pipes.									
Diameter of pipe in inches, .	1	1¼	1½	1¾	2	2¼	2½	2¾	3
Thickness, L.S.G., . .	10	10	10	9	9	8	8	7	7

Feed discharge pipes to be as steam pipes for 30 per cent. higher pressure; but in no case to be taken lower than 125 lbs.

Receiver pipes to be as steam pipes for half the test pressure of the cylinder to which they lead steam; but in no case to be taken lower than 50 lbs.

The above gauges refer to straight pipes only; bends to be suitably strengthened.

It is a very common practice to make pipes of the same thickness, whatever be their size, for the same service. No doubt, for large engines, these thicknesses are not excessive; but when a pipe 2 inches in diameter, to withstand the same pressure as one of 8 inches, is made of the same thickness, there seems an absence of reason.

The Admiralty specifications allow of steel pipes being used very freely for steam pipes, the smaller ones being solid drawn, and the larger ones welded and fitted with a riveted cover strap on the weld outside. The flanges are generally riveted on.

In the merchant service cast iron is used for large pipes conveying water and exhaust steam, and occasionally for the main steam and feed pipes, and where cost and stiffness are of more consideration than weight, they may be used with advantage, but although there seems an element of danger in them, experience has proved them to be reliable and safe with even high pressures of steam. With the higher steam pressures, however, this material has been dropped for steam and feed pipes, and solid drawn steel or welded wrought iron substituted. The flanges are sometimes riveted and sometimes brazed or screwed. Messrs. Howell, of Sheffield, have made some pipes with flanges in one piece by an ingenious system of press.

The joints of all pipes should be in such positions that they are ea access.

Auxiliary Machinery.—Most ships nowadays (both large and small) besides donkey pumps, other auxiliary engines requiring the attention engineer, although some of them are never placed in the engine-roon others (although better there) are not always so placed.

Electric Lighting is most suitable for shipboard as being both conveni to fit and economical in upkeep. As with the feed donkey, so with this, i is highly desirable that it give as little trouble and anxiety to the engineer as possible. To this end the dynamos should be direct driven by an enclosed engine having forced lubrication, as introduced and perfected by Messrs. Belliss & Morcom, Birmingham. A two-crank compound engine with one central valve as made by this firm, as shown in fig. 126, is the most suitable for ship purposes up to 250 I.H.P. For larger powers triple-compound engines are made by this firm, and for small ships single-crank compounds with tandem cylinders answer very well. Fig. 126*a* is the open type as made for H.M. service until recently, when the enclosed was adopted. Except in very small ships it is highly desirable to have two sets of engines and dynamos, each of which is capable of giving at least two-thirds of the total output required at any one time. In larger ships even three such sets may, with advantage, be adopted, so that there is always one set in reserve for any contingency. In H.M. ships there are usually two sets quite independent of one another for lighting only; one for fighting service and one for general use.

When power is distributed by electrical means for driving cranes, winches, windlass and capstans, turrets, and auxiliary machinery generally, there should be two sets of generating plant, each capable of giving at least two-thirds the maximum output. For heating purposes electric energy has been used on shipboard and found to be most convenient. If only a moderate amount of heating is required, this method is a satisfactory one, but for very cold climates the same cannot be said, partly because the

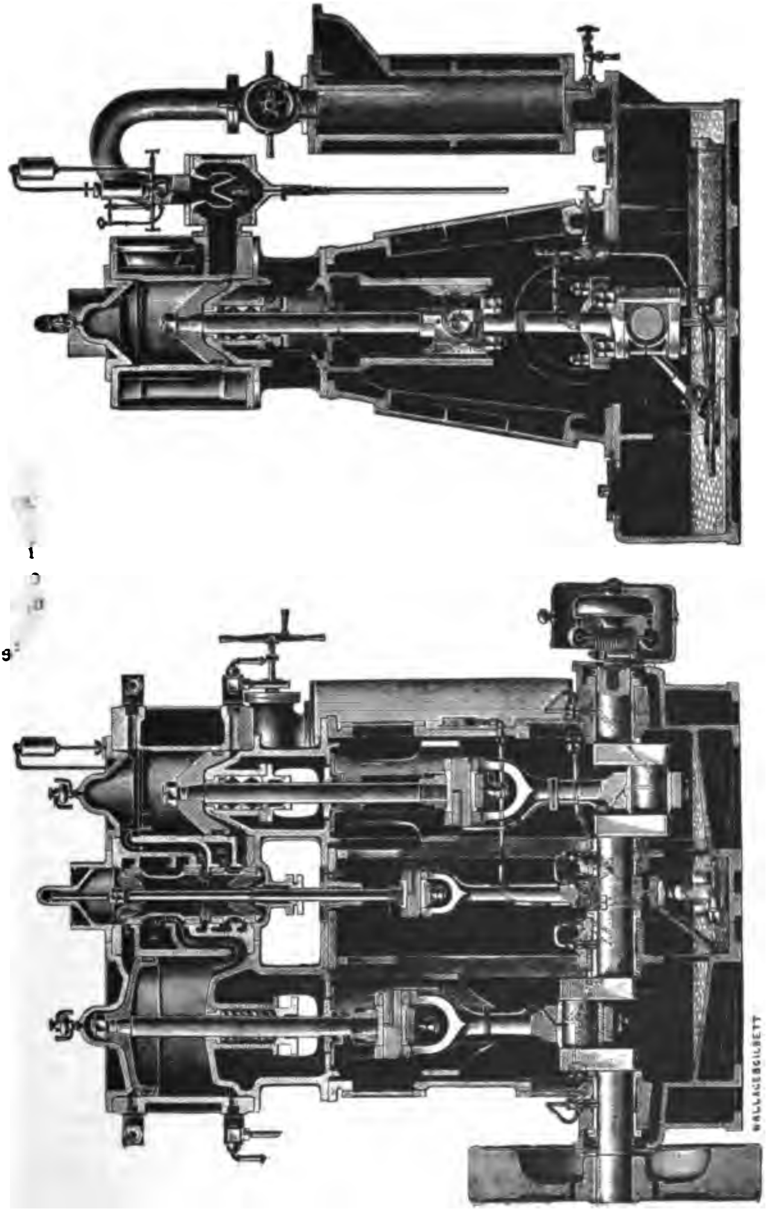

Fig. 126.—Belliss & Morcom's Central Valve Enclosed Compound Engine for Electric Lighting.

expense is great, but chiefly on account of the invisibility of the source of heat affecting the sensibility of those requiring warmth.

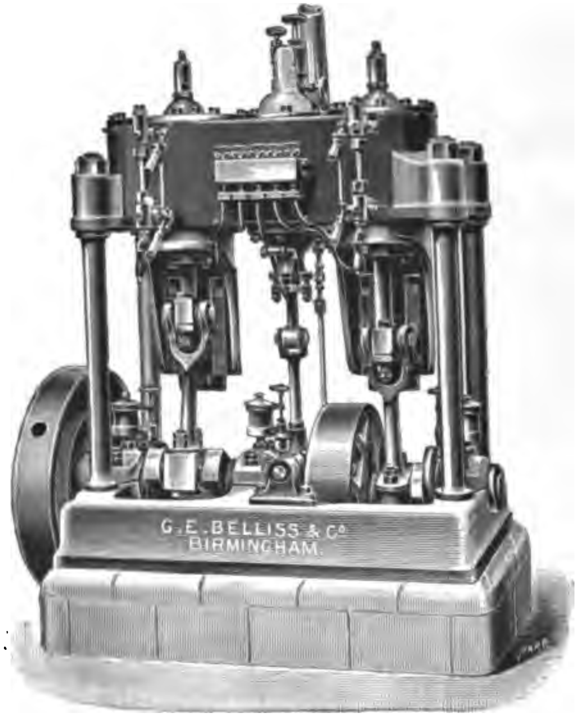

Fig. 126*a*.—Electric Light Engine (Admiralty Open Type Compound).

Steam Steering Gears, now so indispensable on steamships of high power or large size, and so convenient on ships of all sizes, are fitted to even quite small steamers. Fig. 127 is a typical form of the gears now fitted, which have all developed out of that first and most ingenious design of Mr. Macfarlane Gray for steering the "Great Eastern" in 1858. These gears are often fitted in odd places on or about the bridge—sometimes in the chart-room and sometimes in a box to themselves. In the Navy they are now in or near the engine-room, where they can and do receive the attention of greasers on watch. In the mercantile marine they are often, and should be always, similarly situated. *Brown's steering gear* (fig. 127*a*) and telemotor connection to the steering stations differs from all other gears, inasmuch as the engine, &c., is placed on the tiller or quadrant, and moves it by gearing, &c., fitted in the usual way. The differential valve gear is operated by a small hydraulic cylinder piston, piston-rod, &c., which piston gets its motion by pressure from the movement of a similar piston in a cylinder in the steering station, the fluid pressure being conveyed by small copper pipes; the fluid itself is glycerine or a solution of glycerine which will not freeze.

Description of Brown's Patent Steam Tiller.—The importance of fitting steamships, whether of high or low speed, with absolutely reliable steering gear cannot be overrated; but in many cases commercial considerations are allowed to weigh in the design of this vital adjunct to a steamer's equipment, and a large proportion of the damage and losses at sea are attributed to the inefficiency or weakness of such gear. By far the greatest number of steamships have their steering engines placed near the bridge, the communication being made with the quadrant aft by means of chains, rods, or wire ropes, with or without spring buffers to take off the shock of a heavy sea. In conjunction with this, hand gear is fitted aft, having double screws with nuts and cross-head, the mode of connection being by pins dropping into connecting links, or by a clutch working on the rudder-head and engaging the cross-head. The trouble involved in keeping these steering ropes or rods properly adjusted and the various pulleys properly oiled, as well as danger arising from the ropes being carried away, has brought about a change in more recent applications of steering gear. The steering engine is placed aft, being coupled by right- and left-hand screws, and in a variety of other ways, direct to the rudder-head; communication from the steering valve being made by a line of shafting to the bridge, thus dispensing with the objectionable rope or rod communication, which is, in the first-mentioned system, subjected to the full rudder strains.

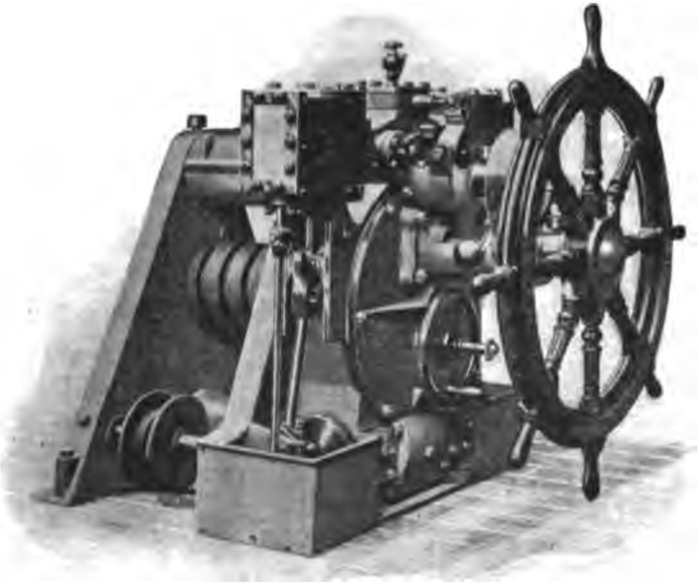

Fig. 127.—Ordinary Merchant Ship Steam Steering Gear (Alley & M'Lellan's).

An ideally perfect steering gear should fulfil the following conditions:—

1. The steering engine should be attached to the rudder-head without the intervention of chains or ropes.

2. It should let go the rudder when unduly strained, and, when the abnormal strain has gone, return automatically to its former position.

3. The connection from steam to hand gear, and *vice versâ*, should be affected without the use of jaw clutches or the shipping of bolts into holes —which operations are difficult to effect when the ship is rolling at sea with the rudder adrift.

4. The communication from the bridge to the machinery aft should be of a kind which dispenses with rods, chains, and shafting, all being equally troublesome to the shipbuilder to arrange, and to the officers of the ship to keep in order.

With reference to condition 3, it is a common practice to fit rudder brakes on ships where clutches are the means of connection; but as simplicity and fewness of parts are of first importance in steering gear, it is better that such a connection between the steering engine and the rudder, or the hand gear and the rudder, should be one which will act both as a clutch and a brake.

To meet these conditions as far as possible, the steam tiller has been designed. In the following drawing (fig. 127*a*) an elevation with hand steering gear and plan are shown.

The prominent feature of this gear, in which it differs from all others, is that advantage is taken of as long a lever as will reach from the rudder-head to the limits of the poop deck, which, in the greatest number of ships, varies from 7 to 10 feet, and in the largest class of vessels has reached the length of 17 feet. It will be obvious that the strains at the end of such a lever will be reduced to the smallest possible amount, and that the gear necessary to give the requisite power to steering the ship will be of the simplest form.

This tiller is shown in fig. 127*a*, A, keyed to the rudder-head, B, and at the other end a jaw, C, is fitted with gun-metal bearings, into which a driving pinion, D, works, gearing into the toothed segment, E, which is bolted securely to the deck. The steering engines are carried on the tiller and move round with it, receiving and exhausting their steam through a double stuffing-box arrangement, F, which also contains the reversing valve, and is mounted on the axis of the rudder-head. The steam cylinders, G, are of the usual well-known construction, fitted with piston valves. Motion is communicated to the pinion, D, which is of cast steel, through the intervention of an expanding friction clutch, H, which is lined with elm wood, and engages the worm-wheel, I. This wheel, to reduce friction, is carefully machined in the teeth, and made an exact fit to the worm, J, which is of Admiralty gun-metal, and works in the worm-wheel without any back-lash or shake. The worm is also curved to the radius of the wheel so that three teeth are engaged, thus prolonging the life of the gearing by three times that of a straight worm.

Motion is given to this worm by the steam engine as shown. The clutch, H, is expanded by a screw bolt and worm-wheel, K, which turns in and out of the nut, L, at one end, the other abutting against a series of laminated springs, M, so that by turning the worm, N, by a handle (provided for the purpose) to the right or left, the steam gear is engaged or disengaged at any position the rudder may be in, and at the same time it forms an efficient brake to seize hold of the rudder in a seaway. In practice it is usual to expand this friction brake or clutch sufficiently tight to put the rudder hard over at full-speed trials; but the springs in any case have not sufficient force to hold the connection tight enough to cause fracture of any part of the machinery. In the event of a heavy sea striking the rudder, it immediately slips, allowing the rudder to move out of position; but by that act the steam valve is opened and the engines bring the rudder back to its normal place. As the steam tiller is intended to work (and in most cases has been so fitted) on the open deck, without any house, the whole of the machinery is placed in a water-tight casing, which forms the framework of the steering engines, access to which is got by the doors, O O. The oiling of the various parts is effected automatically by two valveless oil pumps, P P, driven off the valve rods of the engine. These throw the oil from a well in the bottom of the casing through the hollow piston-rod into the reservoir, Q, and from there a copious supply of oil is supplied to

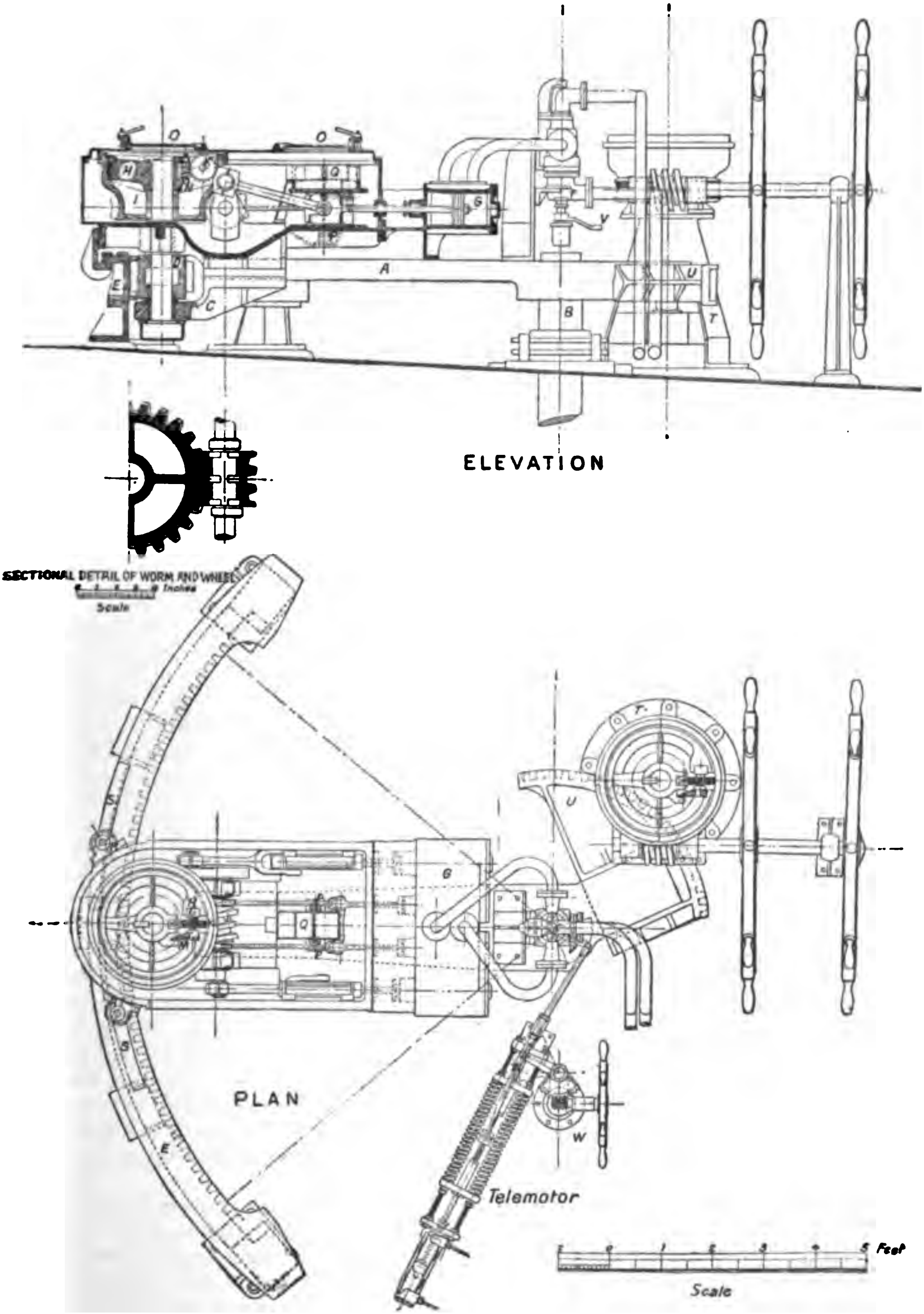

Fig. 127a.—Brown's Steam Steering Gear.

every working part, as well as the piston and valve-rods. In actual practice the oil is renewed once in three months, about 2 gallons being required.

The pinion end of the tiller is carried up by gun-metal slippers and spiral springs under the lugs, R R, which are capable of adjustment. As there is always a tendency of the tiller or quadrant to shake or chatter when there is no strain on, the rudder being fore and aft, the slippers in that position work upon inclined planes, S S, which gives sufficient brake action to prevent any vibration. At the same time, when putting the helm over to such an extent that the pinion bears hard on the rack, the slippers run down on to the flat part of the toothed segment, when the springs slack off and only carry up the weight of that end of the tiller.

The hand gear consists of a strong standard, T, bolted to the deck, and carrying an exactly similar worm-wheel, and worm with hand-wheels and friction clutch as that described in the steam gear. At the lower end of the shaft there is a similar pinion to D, which engages the toothed segment, U, of cast steel, which is securely bolted to the steam tiller. The operation in changing from hand to steam or steam to hand by means of these clutch brakes can be, and has been, performed, without any undue haste, in half a minute. It may here be pointed out that the result of actual experience is that, with this system of hand gear, the friction is one-third of that of the double screw system with nuts and connecting-rods to a cross-head on the rudder-head. Therefore one man on the worm-wheel gear is as effective as three on the double screws.

The hand wheels, it will be observed, are set to one side of the centre line, which economises space fore and aft, and brings the position of the man steering immediately opposite the compass.

The steering valve is operated by the lever, V, which causes the piston valve to turn on its axis inside the trunnion casing, and as the tiller moves round it carries the valve face with it and so closes the port. The lever, V, is connected to the motor cylinder of the telemotor gear, leading up to the bridge by two pipes $\frac{3}{8}$ inch in diameter. In case of accident to these pipe communications to the bridge, a steering station, W, is shown aft, which can be connected to the steering valve.

It is claimed for this design of steering gear that it has the fewest number of parts possible—namely, one pinion, one worm-wheel and worm—which, it can easily be seen, is due to the fact that the toothed segment represents in a 10-feet tiller a steering wheel 20 feet in diameter, and this rack being shrouded to the points of the teeth and bolted at short intervals to the steel deck, is extremely secure.

From a commercial point of view, there is a distinct saving in the adoption of such a design, as no space is required for a steering engine amidships, and no house is required aft—a few stanchions and rods or netting being placed at the extremity of the gear for the protection of passengers.

Auxiliary Condenser.—In these days of high pressures of steam when fresh water only must be used in steam generators, it is essential to save all the steam possible, and as large quantities are used by the auxiliary machinery in modern steamers, it is worth while having a special condenser for this purpose. In the mercantile marine it is generally called a *winch condenser*, as primarily it was fitted to take the exhaust steam from the winches and return it to the auxiliary or donkey boiler supplying them. These condensers in the Navy are similar in design and construction to the main condenser, but of course much smaller; a combined air and circulating pump is attached and kept working, when in use, at a constant speed; an automatic regulating valve can be fitted in such a way that as the vacuum improves steam is reduced, and, when 24 inches of vacuum is obtained, the engine goes "dead" slow.

Fig. 128 gives a good arrangement for the mercantile marine which may be placed in any convenient position in engine or boiler room, as it is self-contained. Sometimes in the mercantile marine a simple cylindrical condenser, as in fig. 128, is fitted so that the ballast pump supplies circulating water, and one of the auxiliary feed-pumps draws away the condensed water; in this case little or no vacuum is maintained, but no water is lost. In every-day work the winch and other auxiliary engine glands and drain cocks leak so badly that only a very poor vacuum is possible with even a good-sized air-pump.

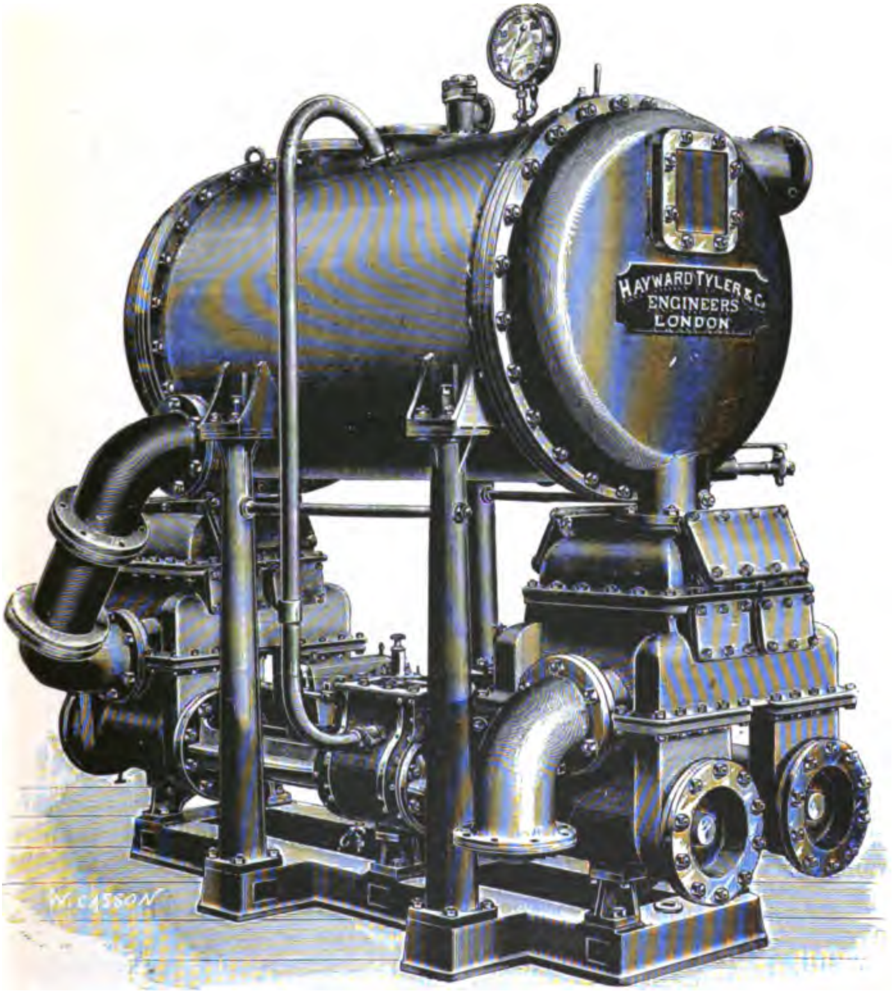

Fig. 128.—Winch Condenser, complete, with Air and Circulating Pumps.

CHAPTER XVII.

BOILERS, FUEL, ETC., EVAPORATION.

THE boiler proper consists essentially of two parts; the one the fire-place in which the fuel is burnt and heat generated, the other in which the heat is applied to boil the water and convert it into steam.

The part where the fuel is burnt is called the *furnace*, and that on which it is laid is called the *grate*.

The quantity of steam generated will depend, in the first place, on the quality of the fuel, and on the quantity burnt; and in the second place on the capacity of the boiler for absorbing and transmitting the heat generated. The quantity of fuel consumed depends on the area of grate, and the draught or flow of air into the furnace.

The *efficiency* of the furnace depends on its capability of burning with as little waste as possible the whole of the fuel in it, and that without superfluity of air; also, to produce perfect combustion there must be as little waste of heat as possible in obtaining the necessary draught. The portion of heat generated which is applied to the production of steam is called the *available heat*.

Every-day experience proves that a grate may consume a large quantity of fuel without thoroughly burning it, and that even when the fuel is thoroughly burnt only a comparatively small portion of its heat may be usefully employed.

The chief loss is at the chimney, which is a rough and ready way of inducing the air to flow into the furnaces with sufficient velocity to cause the fuel to burn; but it is an exceedingly wasteful one, and will, some day, undoubtedly be superseded by a more scientific and economical apparatus. One pound of fairly good coal *can be made* to evaporate 14 to 15 lbs. of water; but in the best of marine boilers only 10 lbs. of water per pound of such coal are evaporated in practice with every care taken, showing that the efficiency of the boiler is less than 0·7 from this point of view. With picked Welsh coal and most careful stoking 12 lbs. can be evaporated if the rate of combustion and evaporation is not high.

Efficiency of the Furnace.—Loss may take place at this part of the boiler from the following causes:—

(1) *Bad stoking*, whereby fuel is lost by falling through the bars only partly consumed, and is thrown away with the ashes. This generally takes place with good fuel from its friability; but may, in some cases, be due to carelessness in laying the fire-bars. Too much cannot be said of bad stoking, as from this cause alone the best-designed and made boiler may prove most inefficient.

(2) *Want of air*, whereby the fuel is not wholly consumed, but part of it passes off through the funnel as carbonic oxide, part in the form of smoke, part is deposited as soot in the tubes, and part is burnt in the smoke-box. This is sometimes due to bad design and want of proper means of

regulating the supply of air; but it is more frequently due to bad stoking and carelessness on the part of the fireman in using the means provided.

(3) *Excess of air*, whereby some of the heat is employed in raising the temperature of the superfluous air, and thereby doing no good. This is generally caused by bad stoking, the bars being uncovered in parts of the grate, and so admitting a too large inflow of air.

(4) *Radiation* through the mouth when the doors are open for firing, and any radiation through openings for other purposes. This cannot be avoided, but should be comparatively small.

The first three of these sources of loss cannot be said to be unavoidable, as with ordinary care the loss from them should be very small compared with the chief loss.

Chimney Draught.—To obtain an efficient draught in a chimney or funnel, the temperature at its base should be about that of melting lead, or nearly 600°. Professor Rankine states that "the best chimney draught takes place when the *absolute* temperature of the gas in the chimney is to that of the external air as 25 to 12."

That is, if T be temperature of the air, then

$$\text{Absolute temperature at the base of funnel} = \frac{25}{12} \times (461^\circ + \text{T}),$$

or temperature according to thermometer (Fahrenheit),

$$= \frac{25}{12}(461^\circ + \text{T}) - 461^\circ = 2{\cdot}08\,\text{T} + 500^\circ.$$

Taking the temperature at 60°, the proper heat at the base of the funnel is then $2{\cdot}08 \times 60 + 500$, or 625°. Now, the heat in a furnace having a natural draught is usually about 2400°, and if the heat at the exit from the boiler to obtain the necessary draught is 600°, it follows that 25 per cent. of the total heat of combustion is wasted owing to this. If, instead of a chimney, a draught were produced artificially, say by means of a blowing engine or fan, a very considerable portion would be saved.

Fuel.—The usual fuels used on board ship are *coal* of various descriptions and qualities, *wood* in those parts of the world where it is plentiful and coal scarcer, and *patent fuels* or combinations of coal with other substances. *Coke* is seldom used, and *oil*, although comparatively cheap, and in some instances most convenient, has not come into general use, notwithstanding a very strong advocacy of it by influential persons. Mineral oil is capable of producing a large quantity of heat, is clean and convenient for stowing, and therefore suitable for yachts and passenger steamers; arrangements are now being made for the regular supply of a high flash point cheap oil at foreign and home ports, so that ships can depend on getting their tanks filled as surely as they can their bunkers. Creosote waste, as well as petroleum waste, can be burnt quite safely and readily in the furnaces of a boiler by means of special apparatus; the one invented by Mr. James Holden, and used by him so successfully for many years in the locomotives of the G.E.R. Co., London, is shown in detail in fig. 129. Refuse from the petroleum works in South Russia, called *astatki*, has for very many years been used on the steamers of the Caspian and Volga for fuel.

Of coals there are several distinct kinds, and many more qualities. There are five distinct varieties, known as—

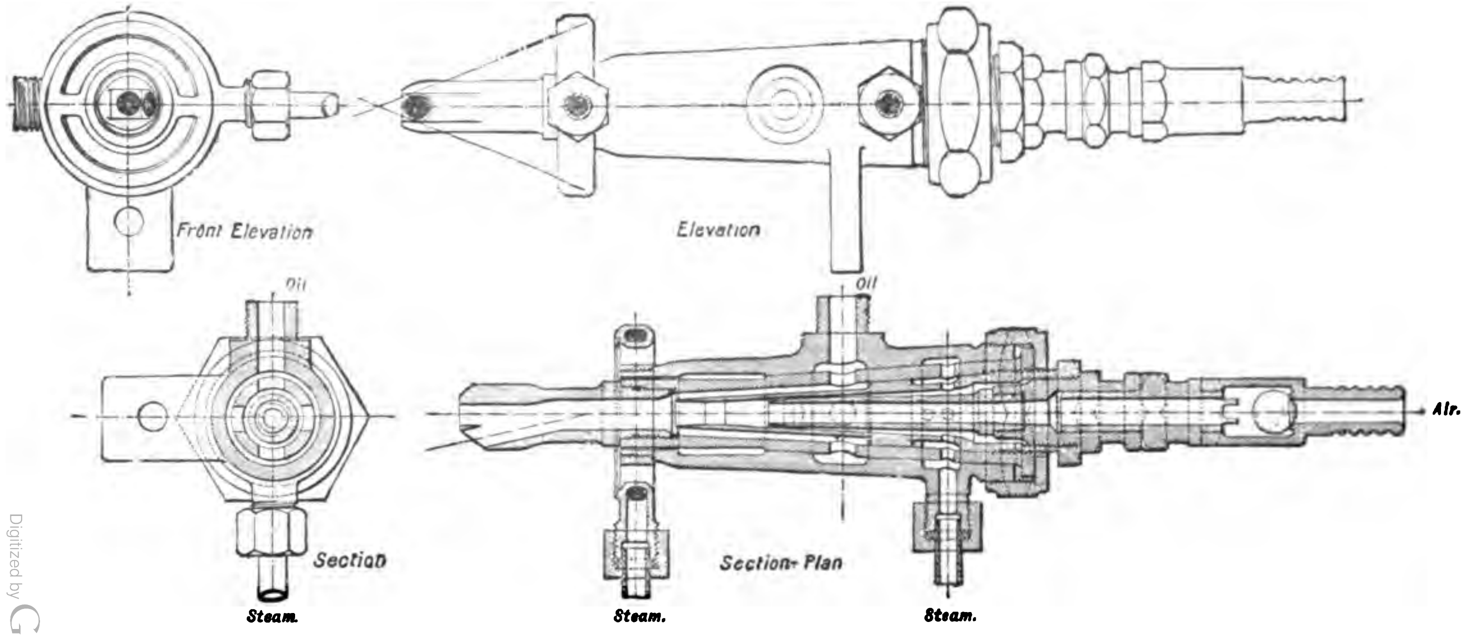

Fig. 129.—Holden's Apparatus for using Oil as Fuel in Locomotives.

(1) *Anthracite*, consisting almost entirely of free carbon, generally jet black in appearance, but sometimes greyish like black lead, has a specific gravity generally of about 1·5, but sometimes as high as 1·9; it burns without emitting flame or smoke, but requires a strong draught to burn at all. It is capable of evaporating (theoretically) nearly 16 times its weight of water, but to obtain good results from it careful stoking is necessary, as when suddenly exposed to heat it is very friable, breaks up into small pieces, and falls through the bar-spaces if disturbed much, as it does not cake. The fires should be worked light when using it, and the coal carefully spread. The heat is very intense and local, so that furnaces intended to burn it should be high in the crowns and protected at the sides by bricks, fireclay, &c., or else have no air-spaces down the sides.

(2) *Dry bituminous coal* contains from 70 to 80 per cent. of carbon, and about 15 per cent. of volatilisable matter; its specific gravity is from 1·3 to 1·45. It burns easily and swells considerably while being converted into coke. The harder kinds do not burn so readily, nor do the pieces stick together so easily when burning, and are generally better adapted for marine boilers.

(3) *Bituminous caking coal*, containing from 50 to 60 per cent. of carbon, is generally of about the same specific gravity as the dry bituminous; it contains, however, as much as 30 per cent. of volatilisable matter, and consequently develops hydrocarbon gases; it burns with a long flame, and sticks together in caking, so as to lose all trace of the original forms of the pieces. It requires special means to prevent smoke.

(4) *Cannel coal*, or *long flaming coal.*—This is seldom used for steam purposes, as it gives off large quantities of smoke, and is very scarce. It is the best coal for the manufacture of gas.

(5) *Lignite*, or *brown coal*, is of later formation than the other coals, and in some instances approaches to a peaty nature. It contains, however, when good, from 56 to 76 per cent. of carbon, and has a specific gravity from 1·20 to 1·35. It also contains large quantities of oxygen, and a small quantity of hydrogen. The commoner kinds of lignite are poor, and contain as little as 27 per cent. of carbon, and therefore are not suitable for steaming purposes.

In some parts of the world, where coal cannot be easily obtained, wood is used for fuel, and the furnace specially constructed to burn it; it contains, on the average, when dry, about 50 per cent. of carbon, 41 of oxygen, and 6 of hydrogen.

Patent Fuels consist principally of coal, and their value depends therefore on the quality of the coal from which they are made. To utilise the small coal from the mines and yards was a somewhat difficult problem, as it could not be conveniently transported, and is difficult to burn in an ordinary furnace. By mixing a small quantity of pitch or tar with it, and baking the mixture in moulds, a hard brick is produced, which is easily handled and burns well. There are many other ways of manufacturing patent fuel from small coal, but the result is the same—viz., a hard brick.

The Value of a Fuel is determined by its chemical composition. All fuels contain more or less of carbon, must have also hydrogen and oxygen in various proportions, and some small quantities of nitrogen, sulphur, &c.

These substances are usually designated by their chemical symbols, that is by the initial letter of the name. They combine in certain fixed quantities, called their *chemical equivalents*. Thus—

Carbon,	symbol	C,	chemical equivalent	12.
Hydrogen,	„	H,	„ „	1.
Oxygen,	„	O,	„ „	16.
Nitrogen,	„	N,	„ „	14.
Sulphur,	„	S,	„ „	32.

A pound of carbon is capable of developing, during combustion, a certain quantity of heat, called the *total heat of combustion*, and is measured by *units of heat*.

The British standard unit of heat is defined as *that quantity of heat which will raise one pound of pure water one degree Fahrenheit in temperature.*

The total heat of combustion of one pound of hydrogen is measured in the same way, and it is also found that the *total heat of combustion of any compound of carbon and hydrogen, is the sum of the quantities of heat which the hydrogen and carbon contained in it would produce if burnt separately.*

If a fuel contains oxygen as well as hydrogen, it is known that eight parts by weight of the former unite with one of the latter to form water, which exists as such in the fuel, and *does not add to the total heat of combustion.* If there is, however, an excess of hydrogen beyond what is required to form with the oxygen the water, the *remaining hydrogen does add to the total heat of combustion*, and may be reckoned in estimating its value.

Hydrogen gas requires 8 pounds of oxygen, and consequently 36 pounds of air to consume it; its total heat of combustion is 62,032 units of heat, and therefore it can evaporate 64 pounds of water from and at 212°.

Carbon, when fully burnt, requires only 2·7 pounds of oxygen, or 12 pounds of air to consume it—that is, to convert it into carbonic acid: its total heat of combustion is 14,500 units of heat, and can therefore evaporate 15 pounds of water from and at 212°. If, however, it is only partially burnt or turned merely into carbonic oxide, half the quantity of air is consumed, and the total heat of combustion is only 4400 units of heat.

Sulphur exists only in small quantities in good coal, and the total heat of combustion is only about 4000 units.

From this information, the following rule is deduced for the total heat of combustion for substances containing carbon, hydrogen, and oxygen.

Total heat of combustion of 1 pound of fuel

$$= 14{,}500\left\{C + 4{\cdot}28\left(H - \frac{O}{8}\right)\right\} \qquad - \quad - \qquad (1)$$

and

Theoretical evaporative power of 1 pound of fuel

$$= 15\left\{C + 4{\cdot}28\left(H - \frac{O}{8}\right)\right\}.$$

C being the weight of carbon, H that of hydrogen, and O that of oxygen, all expressed in fractions of a pound.

Example.—To find the evaporative power of a pound of Welsh coal, whose constituents are, carbon 85, hydrogen 4, oxygen 2·5, others 8·5.

$$E = 15\{0{\cdot}85 + 4{\cdot}28(0{\cdot}04 - \tfrac{0{\cdot}025}{8})\}$$

$$15{\cdot}125 \text{ pounds.}$$

The quantity of air required to burn a pound of fuel may also be estimated in a somewhat similar way.

TABLE XXIV.—FUELS.

Description.	Carbon.	Hydrogen.	Oxygen.	Sulphur.	Ash, &c., including Nitrogen.	Total Heat of Evaporation.	Evaporative Power from and at 212°.	One Ton occupies in Cubic Feet.
						Units.	Pounds.	
Welsh—Ebbw Vale,	87·78	5·15	0·39	1·02	3·66	16,221	16·79	...
,, Powell's Duffryn,	88·26	4·66	0·60	1·77	4·71	15,788	16·34	...
,, Llangeunech,	84·97	4·26	3·50	0·42	6·85	14,682	15·20	...
,, Græigola,	84·87	3·84	7·19	0·45	1·91	14,130	14·63	...
,, Average (38 samples),	83·87	4·79	4·15	1·43	5·89	14,858	15·52	42·7
Newcastle Average (18 samples),	82·12	5·31	5·69	1·24	5·12	14,820	15·32	45·3
Derbyshire Average (7 samples),	79·68	4·94	10·28	1·01	4·06	13,860	14·34	47·4
South Yorkshire,	81·88	4·83	7·47	0·54	2·95	14,296	14·71	46·0
Lancashire Average (28 samples),	77·90	5·32	9·53	1·44	6·18	13,918	14·56	45·2
Scotch Average (8 samples),	78·53	5·61	9·69	1·11	5·03	14,164	14·65	42·0
Irish Anthracite,	80·03	2·30	...	6·76	11·03	13,302	14·5	35·7
American Anthracite,	88·54	2·43	...	0·04	8·60	...	...	42·35
,, Bituminous,	73·21	...	...	0·42	11·27	...	...	42·44
French Anthracite,	86·17	2·67	2·85	...	8·56	14,038	14·53	40·00
,, Hard Bituminous,	88·56	4·88	4·38	...	2·19	15,525	16·10	42·75
,, Caking ,,	87·73	5·08	5·65	...	1·54	15,422	16·00	42·75
Chilian Coal,	63·56	5·43	14·84	2·50	14·13	11,030	11·68	—
Indian ,, Average,	70·20	...	...	...	22·9	...	—	...
Patent Fuel—Warlichs,	90·02	5·56	...	1·62	...	16,495	17·07	...
,, Average,	83·40	4·97	2·79	1·26	6·01	15,000	15·66	34·4
Lignite—Russian,	73·72	6·09	20·19	ash neglected		14,263	14·7	...
,, Poor kinds,	66·51	4·72	28·77	,,	,,	11,444	12·0	...
Coke—Best Durham,	85 to 92	...	...	0·25 to 2·0	4 to 12	12,832	13·30	...
Woods—Beech,	49·36	6·01	42·69	...	1·91			94·4
Oak,	49·64	5·92	41·16	...	3·26	average		94·4
Birch,	50·20	6·20	41·62	...	1·96	when dry		106·1
Fir,	51·79	6·28	41·93	...	...	7,800	8·1	...
Willow,	49·96	5·96	39·56	...	4·33	...	...	124·7
Peat—Fairly dry,	59·6	5·80	29·6	0·3	4·7	9,951	10·30	...
Petroleum—American crude,	84·7	13·1	2·2	...	...	20,240	20·95	39 to 46
,, Refuse,	87·1	11·7	1·2	...	...	19,832	20·53	40
,, Caucasian crude,	86·6	12·3	1·1	...	...	20,138	20·85	42

Weight of air required to *burn* 1 pound of fuel

$$= 12\,C + 36\left(H - \frac{O}{8}\right).$$

To *burn* ordinary coal or coke, 12 pounds of air are required on the average.

To provide for the *dilution* of the gaseous products so that free access is given to the air to reach the fuel, much more is required; so that *to consume one pound of fuel in practice,* 24 *pounds of air are often used.*

At the temperature of 62°, the volume of 1 pound of air is 13·14 cubic feet; therefore, to consume 1 pound of coal or coke 315 cubic feet of air are necessary. If, however, the draught is very good, such as found with artificial means, 250 cubic feet are sufficient.

Professor Kennedy found that the best results were obtained when 18·14 pounds of air were used, and noted the following:—

Air per lb. of coal, ... lbs.	14·5	17·4	17·8	18·14
Loss per cent., ...	9·2	5·0	3·6	0·5

Table xxiv. gives the composition, total heat of combustion, and evaporative power of the various fuels.

Rate of Combustion.—The quantity of coal burnt on a square foot of grate depends partly on its nature, but principally on the draught. Some hard coals, like anthracite, require a very strong draught to burn at all, and some qualities of anthracite burn slowly even then. Bituminous coal burns much more freely than anthracite, and some of the softer kinds consume very rapidly. All coal burns very much more rapidly with a strong draught, as might be supposed, and for that reason, when only a comparatively small boiler can be fitted to supply steam, artificial draught is a necessity. It has also been stated that when the draught is very strong, *a smaller weight of air suffices* to complete combustion.

Artificial Draught.—There are four general methods of causing an artificial draught: (1) The steam blast, by which the products of combustion are ejected from the funnel in the same way that the exhaust steam from a locomotive produces a draught; (2) by an air-blast delivered under the fires with a closed ash-pit; (3) by making the boiler room air-tight, and forcing air into it by a fan, until the pressure is above that of the atmosphere, the only vent being through the furnaces to the funnel; and, (4), by a fan placed near the base of the funnel which draws from the smoke-box and delivers into the funnel; this is known as the *induced draught system.*

The first of these methods is the older and commoner form; it is not very effective, and by no means economical, as there is loss of both coal and fresh water; it also quickly wears out the funnel; but it has the merit of being cheap in first cost, and does well enough if it is only required to be used occasionally to quicken the fires in getting up steam, or during a hot day when steaming with a fair wind, or no wind at all. The second plan has the merit of simplicity, and has been made by Mr. James Howden and others very effective by their special systems. There is no danger with it, and it may be applied to any kind of ship. The third plan is one which is universally adopted in warships, and also in some large and small merchant ships. It is a costly plan, and necessitates a complete change in the ship arrangements in the boiler compartment. It is undoubtedly a most

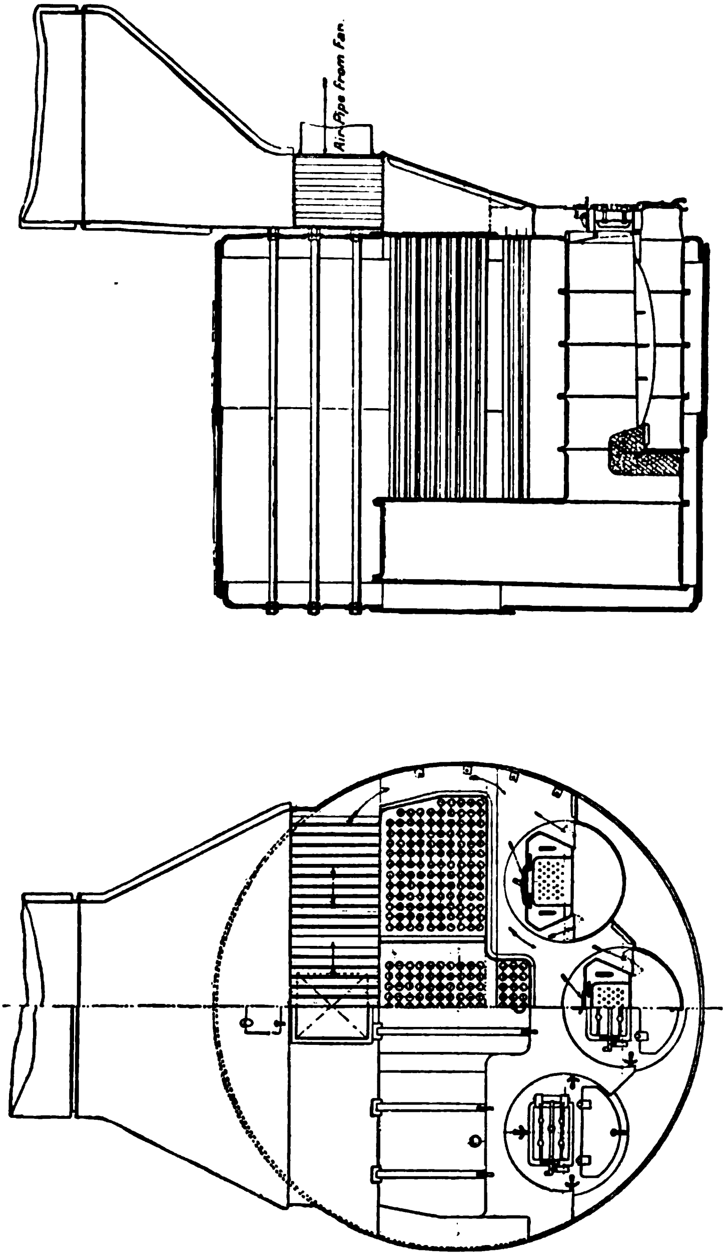

Fig. 129a.—Howden's System of Forced Draught.

efficient plan of artificial draught, but the cost must enter into the calculation of *practical* efficiency, as must also the risk run. The firemen are practically imprisoned in the stokehole, as access to, and egress from it are through an air-tight lock, and when a panic arises the result under such circumstances is often most serious. The fourth plan is simple and less costly, and it is claimed to be less trying to the tubes, &c., but why is not obvious. It has been tried in both the Navy and merchant service. In the latter the Ellis-Eaves patent system, which includes an eduction fan, has been tried with success; but the fan must be most carefully designed and fitted to enable it to deal with gases at 600° F. successfully.

Howden's System of Forced Draught.—The second method of artificial draught above mentioned has been elaborated and perfected by Mr. James Howden, who was one of the first to take up the question of forced draught, and persevere both in experimenting and in educating the public mind, until to-day, when his system has been largely adopted in the Mercantile Marine of all nations with more or less beneficial results. Fig. 129 shows his arrangement whereby the air is supplied to the furnaces by conduits at the sides of the smoke-boxes after having passed through a chamber containing a system of vertical tubes through which the hot air and gases from the smoke-box flow to the funnel. The supply of air is kept up and regulated by means of a fan, and is usually delivered at a pressure equal to half an inch to one inch of water. This system is, therefore, on the regenerative principle, whereby the waste heat in the chimney is utilised to heat the in-coming air to the furnaces, and although the direct saving by using this waste heat is not very large, owing chiefly to the low specific heat of air, the indirect one is considerable from the fact that the combustion of fuel is much more complete and efficient the nearer the air supply approaches the temperature of combustion. Mr. Howden claims a saving of coal by this system amounting to a considerable percentage, and no doubt this will be so when applied to a boiler whose design and proportions are not of the best, but in every case, it must be frankly admitted, there is a material gain by supplying the air at a high temperature and regulating that supply to suit the conditions prevailing at the time.

Ellis & Eaves' System differs from the Howden system chiefly from the products of combustion being drawn at the funnel base and delivered up to the funnel by a fan after passing through the tubes of a heating chamber similar to Howden's. In this case the inflow of air is through the heating chamber and downwards to the furnaces, induced by the suction of the fan, and hence called the "induced draught" system. This system also differs from Howden's in some details. Among others, it is usually fitted with boilers having Serve tubes of considerable diameter, as against the tubes of small diameter fitted with retarders in Howden's case. This system has been fitted by Messrs. John Brown & Co., of Sheffield, to several ships, as well as to boiler installations on shore, with beneficial results in all cases. It need hardly be said, however, that the suction fan has to be very carefully designed and fitted to avoid damage from the heat of the air that passes through it.

Quantity of Fuel Burnt on the Grate.—With good stoking and the ordinary funnel draught, as much as 20 pounds of coal can be burnt per square foot of grate per hour, and under most favourable circumstances, 22 to 25 pounds. In the mercantile marine, 15 pounds is the average amount burnt on a square foot of grate when working economically, and all calculations for a merchant ship should be based on this; for although 20 pounds can be *consumed* when the fires are forced, or on a very windy day when the draught is good, no grate of ordinary length should be supplied with more than 15 pounds to obtain complete combustion, and this is all that will be *consumed* on a sultry day. On a short grate, say 1·3 to 1·5 the diameter of the furnace in length, 20 pounds of coal may, however, be burnt economically. With moderate forced draught, 20 to 30 pounds can be readily burnt on a square foot of grate, and 30 to 50 pounds with good air pressure either *forced* or *induced*, and with short grates and good stoking even as much as 60 pounds. A locomotive with a draught produced by the exhaust steam can burn, as a rule, 65 pounds of coal per square foot of grate; and the modern express engine burns as much as 80 pounds.

The coal consumed in torpedo boats with the closed stokehole, and a pressure of air *in the stokehole* above that of the atmosphere equal to 6 inches of water, is as much as 96 pounds per square foot of grate, and is even 62 pounds at a pressure of three inches only. Mr. Thornycroft eclipsed this in the yacht "Gitana," where 138 pounds of coal were burnt per square foot of grate; this is exceedingly high, and is only such as can be obtained under the most favourable conditions with every care taken in the ventilating arrangements; the evaporative power under such conditions is by no means high, although, perhaps, higher than could have been anticipated. In the Navy an air pressure of ½ inch is allowed to count as natural draught, and 2 inches is the limit allowed for forced draught in large ships with tank boilers. With those pressures the combustion is very good and evaporative power very fair.

Size of Funnel.—The natural draught, or that obtained by means of a funnel, is very much influenced by the area of its transverse section, and by the height or distance of its top from the level of the fire-bars. The draught in a funnel is due to the difference in density of the column of gas in it from that of the surrounding atmosphere; the contents of the funnel rise upwards, on the same principle that a bubble of air rises through water. The density depends on the temperature of the gases at the funnel base, and for this reason a good draught cannot be obtained without a comparatively high temperature. The good draught observable on a windy day is due to the tendency to form a vacuum at the mouth of the funnel, by the wind blowing across it. Professor Rankine has given the following formulæ for chimney draught:—

Let w be the weight of fuel burned in a given furnace per second in pounds.

V_0 the volume at 32° of the air supplied per pound of fuel.

τ_0 the *absolute* temperature at 32° Fahr., which is 461° + 32°,

τ_1 the absolute temperature of the gas discharged by the chimney, whose sectional area is A; then

Velocity of the current in the chimney in feet per second is

$$= \frac{w \times V_0 \times \tau_1}{A \times \tau_0}.$$

The density of that current in pounds to the cubic foot is very nearly

$$= \frac{\tau_0}{\tau_1}\left(0{\cdot}0807 + \frac{1}{V_0}\right);$$

that is to say, from 0·084 to 0·087 × ($\tau_0 \div \tau_1$).

Let l denote the whole length of the chimney, and of the flue leading to it in feet;

m its "hydraulic mean depth;" that is, its area divided by its perimeter; which, for a square or round flue and chimney, is one quarter of the diameter;

f, a coefficient of friction, whose value for currents of gas moving over sooty surfaces is estimated by Peclet at 0·012;

G, a factor of resistance for a passage of air through the grate, and the layer of fuel above it; whose value, according to the experiments of Peclet on furnaces burning from 20 to 24 pounds of coal per square foot of grate, is 12.

Then according to Peclet's formula,

The *head* required to produce the draught in question is

$$= \frac{\mu^2}{2g}\left(1 + G + \frac{f.l}{m}\right),$$

which, with the values assigned by Peclet to the constants, becomes

$$= \frac{\mu^2}{2g}\left(13 + \frac{0\cdot012 \times l}{m}\right).$$

When the *head* is given the value of μ may be calculated, and then,

Weight of fuel which the furnace is capable of burning *completely* per hour

$$= \frac{\mu \times A \times \tau_0}{V_0 \times \tau_1}.$$

It is usual to reckon the *head* by taking 1 inch of water as the unit; then,

$$\text{Head in inches of water} = 0\cdot192 \times h \times \frac{\tau_0}{\tau_1}\left(0\cdot0807 + \frac{1}{V_0}\right).$$

Mr. Thornycroft found, by careful experiment with steam launches and torpedo boats having "loco" boilers and working with a *plenum* (that is, with a closed stokehole into which air is forced), "that of the initial pressure, the resistance of the tubes accounts for about seven-tenths of the whole, the resistance of the fires and fire-bars being only about one-tenth;" and that "the pressure in the funnel, as measured, was sensibly equal to atmospheric pressure."

Professor Rankine also stated that if H be the height of the funnel, τ_2 the absolute temperature of the external air, then—

Head produced by chimney draught

$$= H\left(0\cdot96\,\frac{\tau_1}{\tau_2} - 1\right).$$

or, taking h as the head,

Height of chimney required to produce a given draught

$$= h \div \left(0\cdot96\,\frac{\tau_1}{\tau_2} - 1\right).$$

The velocity of the gas in the chimney is proportional to $\sqrt{h}$, and therefore to $\sqrt{0\cdot96\,\tau_1 - \tau_2}$.

The density of that gas is proportional to $\frac{1}{\tau_1}$.

The weight discharged per second is proportional to velocity × density, and, therefore, to $\frac{\sqrt{0{\cdot}96}\,\overline{\tau_1 - \tau_2}}{\tau_1}$; which expression becomes a maximum, when $\tau_1 = \frac{25}{12}\tau_2$. Therefore the best chimney draught takes place when the absolute temperature of the gas in the chimney is to that of the external air as 25 to 12.

When this condition is fulfilled $h = H$.

That is, the height of the chimney for the best draught is equal to the *head* expressed in hot gas, and the density of the hot gas is half that of the air.

In practice the size and height of chimney are governed rather by circumstances than by scientific investigation; the diameter is fixed by arbitrary rules based on successful practice, and the height such as suits the appearance or service of the ship.

A convenient rule is $H = {\cdot}007\left(\frac{C}{A}\right)^2$; or $A = \frac{C \times {\cdot}084}{\sqrt{H}}$. H is height in feet; A, the area of section in square feet; and C, the consumption in pounds per hour of fuel on the grates connected to the chimney.

Evaporation.—The heat of the gases from the furnace is to be absorbed by the surfaces with which they come in contact on their passage to the chimney, and the efficiency of this part of the boiler depends on the capability of those surfaces to readily take up the heat, on the material to transmit it by conduction to the inner surface or that with which the water is in contact, and on that inner surface being in such a condition as to give up the heat to the water. The *internal* efficiency of the boiler depends on the convection or circulation of the water in the boiler, whereby fresh portions are successively brought in contact with the hot surfaces. The importance of this latter factor was seldom appreciated in estimating the efficiency of a boiler, although now engineers give it first consideration.

When the furnace is internal—that is, when it forms a part of the boiler proper and is surrounded with water—a large proportion of the total heat of combustion is absorbed by it, partly by direct contact with the hot fuel at the sides, and partly by radiation from the glowing surface of the incandescent fuel when coked. The furnace also absorbs heat from the hot gases passing along its surfaces.

The furnace or fire-box of a locomotive boiler is usually made of copper, and it has been proved by experiment that a very large proportion of the whole heat generated in it is absorbed by it, consequently its evaporative efficiency is very high. The furnace of the ordinary marine boiler is of steel, which is not so good a conductor of heat as copper, and, therefore, does not transmit such a large proportion of the total heat of combustion, although its evaporative power is still high. (For conductivity of metals see Chap. xxv.).

Combustion.—The combustion of the fuel is not always *completed* in the furnace of the marine boiler, as the gases distilled from it during the process of coking escape to the chamber beyond the furnace before sufficient air has been supplied; and also it sometimes happens that the temperature *above the fuel* is not sufficiently high to cause ignition. The carbon often unites in the furnace with only sufficient oxygen to form carbonic oxide, and this flows into the combustion chamber; if, then, a further supply of air is found there, another portion of oxygen is taken up, and carbonic acid gas is formed; that is what usually takes place with bituminous coal, and unless this second supply of air is provided, the combustion of a large portion of the fuel is not completed in the boiler. If the fire is completely covered with *green*—that is, fresh fuel—the part next the hot surfaces parts with its volatile elements, which rise and become cooled in so doing, so that when they come in contact with the oxygen of the air, the temperature is not high enough to cause them to chemically unite—that is, to ignite, and consequently they merely mix mechanically and flow on until they pass out at the mouth of the funnel and appear as smoke.

When, however, by careful stoking and regulating the supply of air, so that combustion is completed in the combustion chamber, the evaporative power of that part of the boiler is high, and with the furnaces is the most valuable part of the boiler for transmitting heat to the water.

Heating Surface.—As has been stated, the efficiency of the heating surface of the boiler depends on the material, its thickness, and on the state of the surfaces in contact with the hot gases and water. The furnaces and combustion chambers are of steel, whose conductivity is inferior to that of copper, but is still good (*vide* Chap. xxv.). It is, however, very materially affected by its internal condition—that is, by any want of homogeneity.

The superior evaporative power of the furnace is due in great measure to the cleanness of the surface exposed to heat; there is no deposit of soot or ash on it, and the smallest possible amount of oxide; the combustion chamber also is generally in the same condition. The roughness of the surface exposed probably increases its power of receiving heat, not so much from any abstract virtue in that state, as from the *actual surface* being greater than if smooth. Very much depends on the condition of the inside surface exposed to the water; if it is quite clean and *smooth*, it is not so efficient as slightly dirty and rough; the best condition being roughness with freedom from coating of bad conductors. If a metallic surface is smooth and clean, evaporation from it is slow and intermittent, because on it is formed a film of steam, which is a bad conductor, and this will only disperse when its buoyancy overcomes the attraction; when the attraction is overcome, the steam rises suddenly *en masse*, the surrounding water then flows into its place, when a film is again formed, and a bubble accumulates as before. If, on the other hand, the surface be rough, the film is broken up by numerous points on the surface, which serve as accumulators and starting points for myriads of small bubbles; these are formed quickly, rise freely and continuously, giving a rapid and steady supply of steam. It is for this reason that many boilers *prime*—that is, work with violent ebullition—when quite new.

Tube Surface.—The tubes of a marine boiler represent the greater portion of the total heating surface, and it is to this part that great attention is required, both in designing and working a boiler. They are usually made of iron in the mercantile marine, and of steel in naval boilers. The conductivity of the brass, which was formerly used in the Navy, is theoretically double that of the iron; but in practice with foul surfaces, there is not so great a difference, their relative value being then about 3 to 2; to obtain, therefore, results commensurate with their extra cost, brass tubes should be kept very clean.

Much stress is sometimes laid on certain experiments which show the relative evaporative power of certain portions of the tubes, to prove thereby that the final foot, or two feet, of tube is practically useless in *every* boiler. It is not very surprising to find that the first foot of tube has a high evaporative power, and that the power decreases rapidly the farther removed the portion is from the furnace or combustion chamber. Considering that the temperature of the gas on entering the tube is 2000° to 2400°, and on leaving it should be not more than 600°, while the temperature of the water surrounding the tube is very nearly the same (386° F.) at every part, the transmission of heat should be very much more at the entering end than at the other; moreover, the end next the funnel is more liable to get dirty from deposit of soot, &c., and so to rapidly fall off in efficiency. But in continuous practice it was not found that there was such a large difference in the value of the two ends of the tubes, because the end at which there is the rapid evaporation soon became covered with scale, and its efficiency reduced when salt-feed water was used; the temperature was then not so much reduced after passing through the first portion, and for this reason the last portion was found to have a higher efficiency than before. The size of tube has also some influence on its efficiency, for whereas the surface increases as the diameter, the contents increase as the square of the diameter. If a tube, then, of 4 inches diameter be substituted for two of 2 inches diameter, the surface is the same, while the cubic capacity is doubled; if the rate of flow be the same in both cases, the 4-inch tube will pass twice the quantity of gas through it that flows through the two 2-inch ones, and have only the same surface to absorb and transmit the heat. If the quantity flowing through the tubes be the same in both cases, the 4-inch tube is still at a disadvantage, inasmuch as the mean distance of the gas from its surface is greater than that in the 2-inch ones. The velocity through the small tubes will, in the latter case, be double that in the 4-inch one, and will therefore cause a brisker circulation of the hot gas, and the liability of soot and ash deposit is considerably reduced. Hence tubes of smaller diameter are used with advantage with forced draught, for the same evaporation is effected with smaller amounts of surface. The "Serve" tube made by J. Brown & Co., having 6 or 8 ribs projecting inside, has the great advantage of large absorbent surface in proportion to its cubic capacity. The twisted strips of iron placed inside the tubes by Mr. Howden, and called "retarders," have the effect of reducing the cubic capacity as well as circulating the hot gases; the retarders, however, can transmit very little of the heat they absorb, as the ribs of the Serve tubes do.

Evaporative Power.—The probable evaporative power of a boiler may be found approximately by the following formula:—

Let e_1 be the *theoretical* evaporative power of the fuel, F the weight of coal burned on the grate in pounds per hour, and K the total heating surface in square feet; then

Pounds of water evaporated per pound of fuel burnt

$$= 1{\cdot}833 \left(\frac{K}{2K + F}\right) e_1.$$

Example.—To find the evaporative power of a boiler which burns a fuel whose theoretical evaporative power is 15, the number of pounds burnt per hour on the grate is 800, and the total heating surface is 1000.

Here

$$E = 1{\cdot}833 \left(\frac{1000}{2000 + 800}\right) 15, \text{ or } 9{\cdot}825 \text{ lbs.}$$

The efficiency of the boiler is, by this rule,

$$1{\cdot}833\left(\frac{K}{2\,K + F}\right), \quad \text{or } 0{\cdot}655.$$

This rule is only approximate, however, because the heating surface is not always wholly *real* heating surface, some being only nominal; many boilers as made have given better results after the removal of tubes, which very materially decreased the nominal heating surface. If a boiler has just enough surface to absorb heat from the gases, so that the temperature at the funnel base is only such as is sufficient to produce the required draught, then the heating surface is effective; any surface added to this is superfluous, and in very many cases does positive harm.

The temperature at uptake depends on the temperature of the water in the boiler; for instance, at 300 lbs. pressure, it will be 422° F.; so it will be no use to try to bring the temperature of the gases below 400°, and with such a temperature there the efficiency of the tubes will be very low at 200 lbs. when the temperature of the water is 387° F. It is therefore manifest that it is better to have a higher temperature in the smoke-box, and reduce at funnel base by air-heaters, feed-heaters, or super-heaters.

CHAPTER XVIII.

BOILERS—DESIGN AND PROPORTIONS.

THERE are four distinct classes of marine boiler now in use, each of which may be subdivided in various ways. These are—

(1) The *cylindrical* boiler with return tubes, made for pressures up to 225 lbs.

(2) The *locomotive* boiler and *gunboat* boiler, having tubes extending in line beyond the fire-box or combustion chamber, made for pressures up to 200 lbs.

(3) The *tubulous* boiler, composed wholly of large straight, or nearly straight, tubes and their connections, made for pressures up to 300 lbs.

(4) The *express* water-tube boiler, having small straight or bent tubes, and made for pressures up to 250 lbs.

The Rectangular Boiler, now in use only in old warships and a few modern paddle steamers, was made in two distinct ways, and known as a *dry-bottom* or *wet-bottom* boiler. The wet-bottom boiler was so called because the furnaces were inside, and independent of the shell, and wholly surrounded with water—that is, the furnaces had wet bottoms.

The dry-bottom boiler differed from this, inasmuch as the sides of contiguous furnaces were united at the bottom, so as to form a water space between them, and each furnace was then devoid of a bottom, except the plate fitted to retain the ashes, &c.

When steam of 30 lbs. pressure only was used, the dry-bottom boiler was generally found in merchant ships, while the wet-bottom boiler was always employed in the Navy. There are still such boilers in existence in the

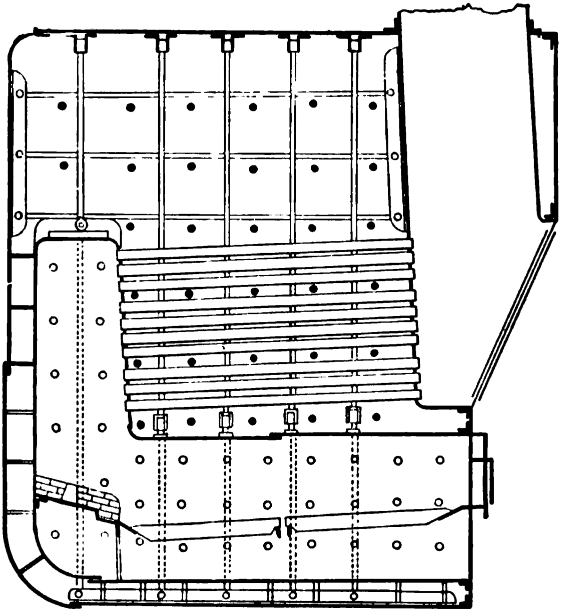

Fig. 129*b*.—Rectangular Boiler.

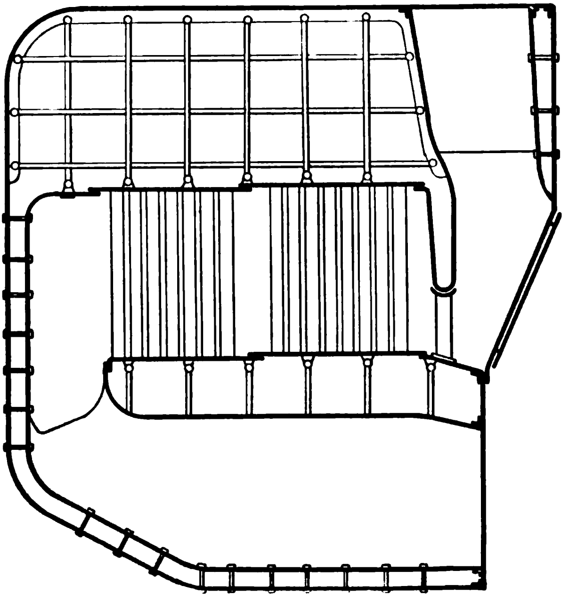

Fig. 130.—Cochran's Boiler.

Navy, but no ship will be fitted with them again, even for such low pressures as they worked at, for a cylindrical boiler is very much cheaper, and can be made to do the work.

Cochran's Boiler.—Fig. 130 differed from the ordinary boiler, having a chamber extending from the combustion-chamber to the smoke-box, in which is a large number of small vertical tubes. This has been called for distinction a water-tube boiler, because in it the water is inside the tubes, and is interesting in the light of some modern inventions, and as occupying a place in the history of the water-tube boiler. It was found by experiment to be more efficient *when clean* than the ordinary boiler, but its efficiency fell off as the tubes became dirty; and as it was most difficult to clean them externally as well as internally, its general efficiency was less than that of a

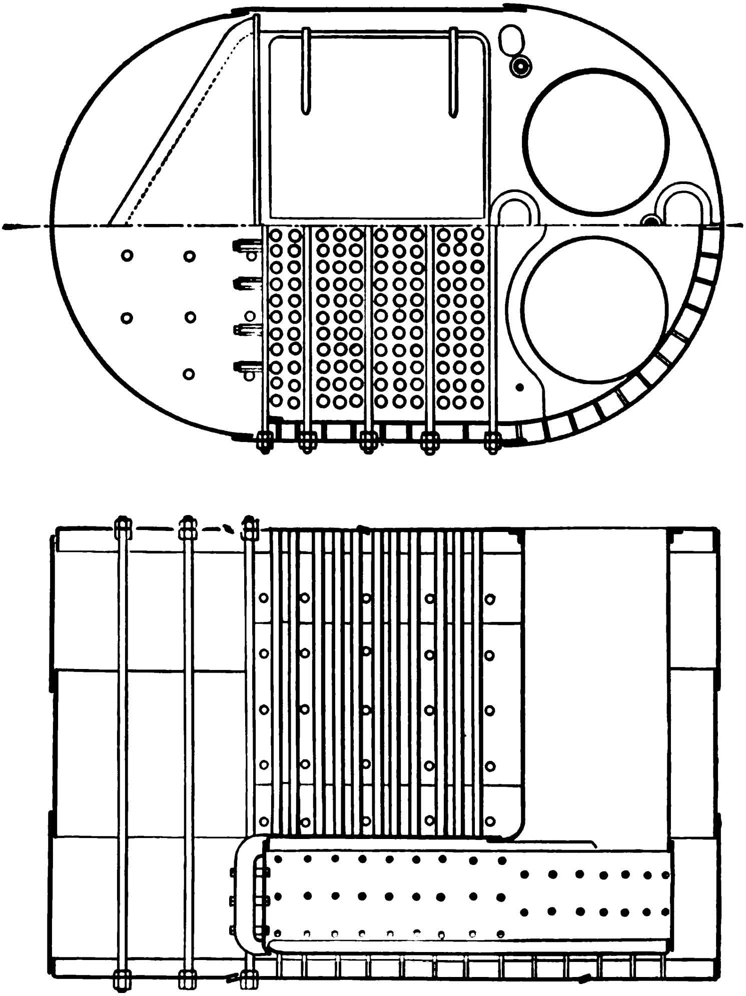

Fig. 131.—Oval Boiler.

well-designed boiler on the ordinary plan. A good test was given to this class of boiler by the Admiralty; H.M.S. "Vanguard" being fitted with one half of the whole number on this plan, and the other half of the ordinary design. It was found that when quite clean the former evaporated about 1 lb. of water per lb. of coal consumed more than did the latter, but after a few hours' work they were equal.

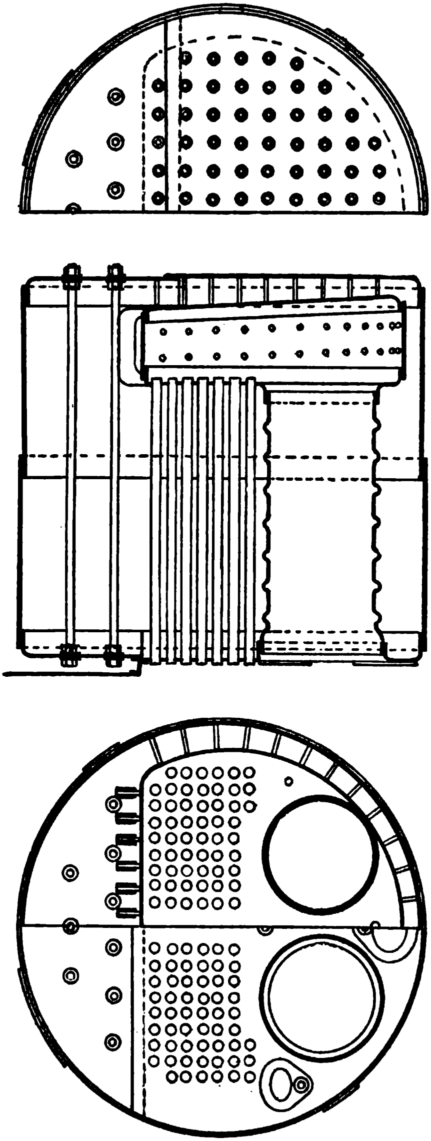

Fig. 132.—Single-ended Cylindrical Boiler (Purves' Furnaces).

Cylindrical Boiler.—To avoid altogether the necessity for vertical stays as well as the transverse horizontal ones, the shell is of necessity a complete cylinder; and, although it was the demand for the higher pressures which became possible after the introduction of the surface condenser that brought the cylindrical boiler into general use, the form was often adopted for pressures as low as 30 lbs. It is lighter, cheaper, and easier to make than the box boiler, and quite as durable when worked under similar conditions.

There are many varieties of cylindrical boilers in use, but they may be divided generally into three classes, viz.:—

(1) *Single-ended*, or single-fired boilers. (Figs. 132, 133).

(2) *Double-ended*, or double-fired boilers. (Figs. 134, 135).

(3) *Oval* boiler. (Fig. 131).

The single-ended boiler has furnaces and tubes only at one end, and is constructed up to as large as 17 feet diameter and 12 feet long. The chief difficulty in designing such large boilers on this plan is to provide adequate grate area for the total heating surface which can be obtained. The number of furnaces cannot well exceed four, and is more generally three in large single-ended boilers. It is true the Admiralty had some cylindrical boilers made in 1854 for a pressure of 60 lbs. with five furnaces, three above and two below, but they were not a success, being most difficult to stoke.

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Small boilers have usually two furnaces, and with this number are more efficient than with three even when of moderate size.

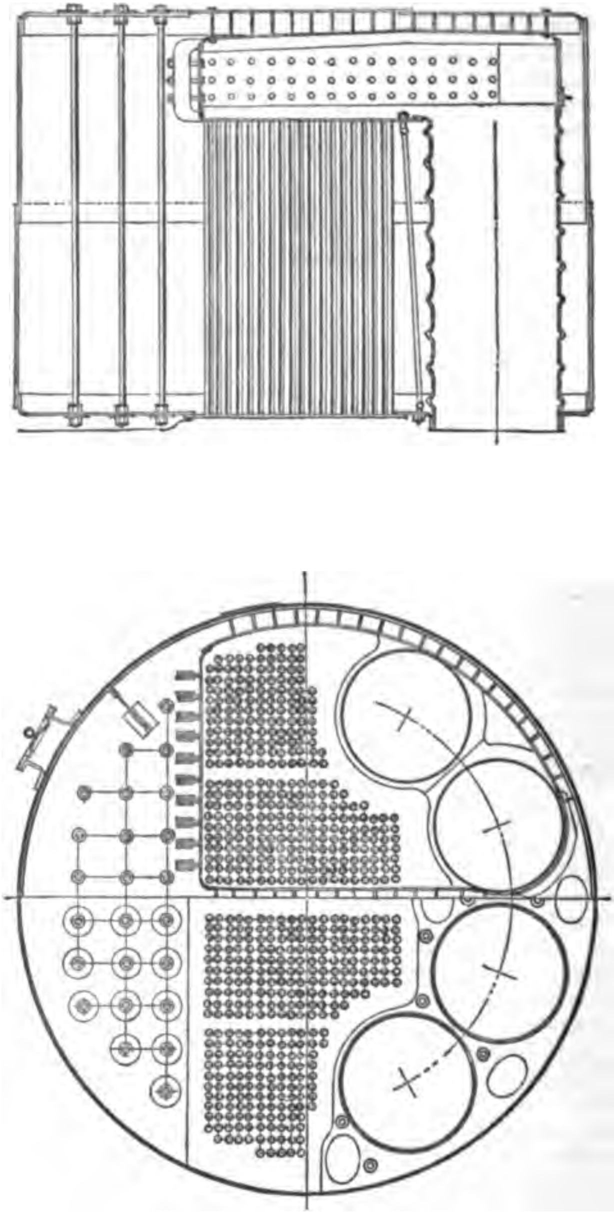

Fig. 133.—Four-Furnace Cylindrical Boiler.

The number and size of the furnaces must, however, depend on the size of the boiler and the heating surface it is to contain. It is found in practice that large furnaces are more efficient as coal consumers than small ones, and the reason is not far to seek. The grate area with the same length of fire-bar increases as the diameter, while the section through which the air passes, both above and below the bars, increases as the square of the diameter; it is also possible to give a good inclination or rake to the bars with a large furnace, which very materially assists combustion. In practice the fire-bars are not of course always of the same length, but they do not increase in length as the furnace does in diameter, and consequently the air passages increase more rapidly than does the grate area when the diameter of furnace is increased. Furnaces should not be less than 36 inches, nor more than 48 inches in diameter, except under special circumstances. Taking this as a rule for guidance, boilers may be made up to 9 feet diameter with one furnace, up to 13 feet 6 inches diameter with two furnaces, up to 15 feet with three furnaces, and beyond that diameter four furnaces are necessary to avoid too long length of grate.

A single-furnace boiler has of course one combustion chamber, a two-furnace boiler may have one chamber common to the two furnaces (fig. 132), or a separate one to each. When there is only one boiler in the ship the latter plan is preferable, as then the bursting of a tube cannot wholly disable the boiler; when there are two or more boilers one chamber common to the two furnaces is preferable, as by stoking the fires alternately an even supply of steam is kept up and the smoke consumed. A three-furnace boiler has usually three separate combustion chambers, and the same remark applies to it as to the two-furnace boiler. The four-furnace boiler has generally only two combustion chambers, one wing and one middle furnace having a common chamber; but some engineers prefer three chambers, the two middle furnaces having one in common, and each wing furnace a separate one.

The chief objection to two large furnaces instead of three smaller, and to three larger ones instead of four smaller, is the longer grate required to get the requisite area, and to the large amount of dead water between the furnaces at the bottom. There is also to be considered the limit placed by the Board of Trade and Registry rules to avoid risk of collapsing by direct crushing of the metal, which often prevents the adoption of the larger furnace with the higher pressures.

It is unusual and certainly most difficult to use plates above 1½ inches thick in the construction of a boiler shell, and it is this consideration which fixes the limit of diameter.

The Double-ended Boiler (fig. 134) has furnaces at both ends with return tubes over them, and is generally tantamount to two single-ended boilers back to back, but with the backs removed. It is made up to 16 feet diameter and as much as 20 feet long; but such very large boilers are unusual, partly owing to the want of facilities for moving such great weight, and partly because the conditions under which such large boilers are possible are limited to very large steamers.

The double-ended boiler is lighter (*vide* Table xxiv.) and cheaper in proportion to the total heating surface than the single-ended boiler, and its evaporative efficiency in practice is generally higher. On the other hand, greater care is necessary in designing and in working it. That it is lighter is obvious, and that it is cheaper may be inferred from the fact that there is less material, and less labour consequent on the reduced quantity of material.

The simplest form of this kind of boiler is one in which all the furnaces open into one common combustion chamber (fig. 134); this form, although at one time common enough, is now seldom adopted. The objections to it are, that the bursting of one tube may disable the whole, that the cleaning of one fire causes the efficiency to sink very low on account of the whole being affected by the inrush of cold air, and that unless special means be provided to promote proper circulation there is a strong tendency to *prime*. Some of these objections are got over by building a thin brick wall with special bricks across the middle. Mr. Howden recommends this arrangement with his system of forced draught, such being necessary whenever there is a chamber common to opposite furnaces.

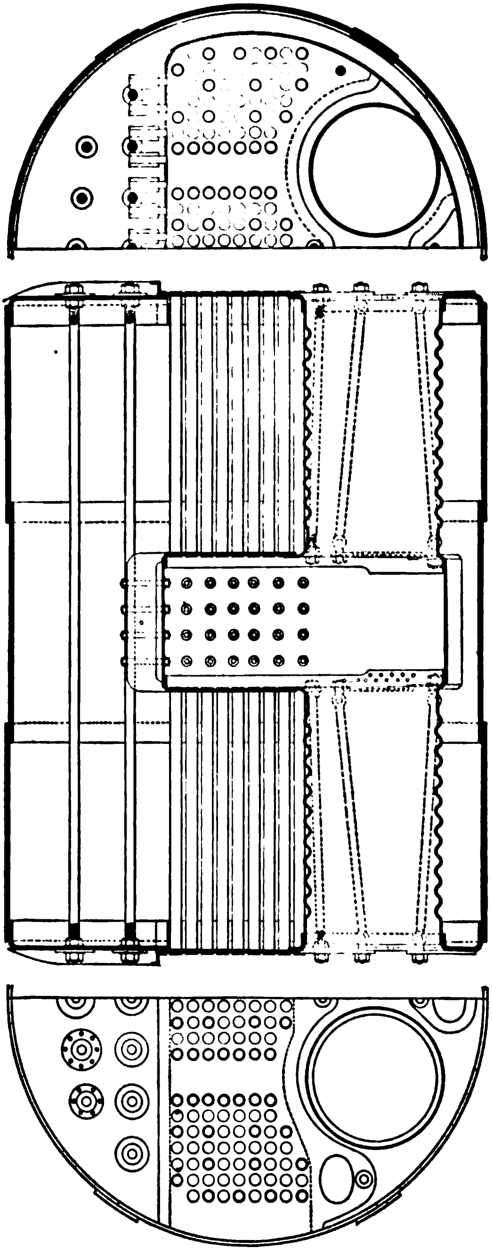

Fig. 134.—Double-ended Boiler with One Combustion Chamber.

The next simplest form is one in which opposite furnaces have a combustion chamber in common (fig. 135) — that is, it differs from the first by having the combustion chamber divided longitudinally by water spaces. This avoids the chief objections raised against the first form while retaining its chief advantages, which are, simplicity of construction, by avoiding the flat back of the combustion chambers, with the necessary stays, &c., and the greatest heating surface within the smallest limits of length. It is often urged against this form of boiler that the tubes are very liable to leakage at their back ends, arising from the rush of cold air against the tube plate when the door of the furnace opposite it is open, causing it to buckle. It sometimes happens that the tubes in this kind of boiler do show a tendency to leak, but it is then generally due to the want of expansion on the part of the first row of stays above the combustion chamber when they are placed too close to the tubes. If these stays are at least 12 inches above the tubes so as not to hold the front tube plates too rigidly, then when steam is being got up the expansion of the tubes simply causes the plates to spring very slightly instead of to start their ends and cause them to leak. The leakage from springing of the tube plate from exposure to cold air can only take place when the combustion chamber is unduly short, and when there is an insufficient number of stays to the tube plates.

This particular form of boiler is very generally used; the evaporative results obtained from it are most satisfactory, and experience does not show it to be liable to more leakage than other boilers. Common care only is required in raising steam, and the opening of fire-doors to check evaporation is a reprehensible practice at all times and for all boilers. A brick semi-partition in the middle of the combustion chamber will prevent the cold air rushing on to the opposite tube plate, and it acts also as an equaliser of

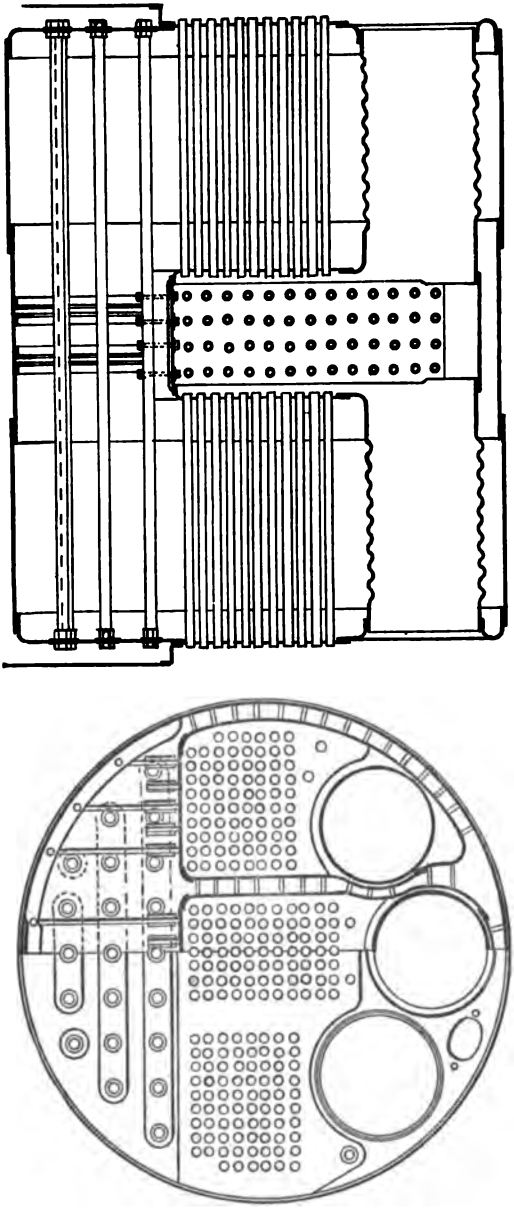

Fig. 135.—Double-ended Boiler (Fox's Furnaces).

temperature in the combustion chamber at all times. If, however, the combustion chamber is too small, this will only magnify the defect by causing intense local heat, and thus tending to crack the plates.

Another common form of double-ended boiler has the furnaces at one end with one chamber common to them, and those at the other end with another chamber in common. The boiler is then longer than either of the other forms, and more expensive; the combustion chambers have large flat backs, requiring a very large number of stays, which prevent their being properly cleaned from scale.

The last form, which is by far the most expensive and heaviest, but is now often adopted, is one in which each furnace has an independent combustion chamber. There is little need of description, as it is to all intents and purposes two single boilers, except that the water and steam are common to the two parts.*

Oval Boilers are included under the generic term of cylindrical, as they partake of the principal features of that class. The transverse section is, however, not an ellipse, but is really formed by two semicircles with a rectangle intervening. The flat sides thus left between the semi-cylinders require staying, the first rows being at the commencement of the flat. There are both single- and double-ended oval boilers, which for pressures under 120 lbs. may be made both simply and economically to very large sizes, as the thickness of shell plate depends on the diameter of the cylindrical part. Two very large furnaces may be thus fitted into a cylindrical part of comparatively small diameter, sufficient heating surface being obtained by giving the requisite height. This form is most convenient when the boilers have to be stowed fore and aft, and the diameter is limited by the breadth of the ship between the stringers, and when forced draught is employed they permit of the large amount of heating surface in proportion to the grate area which is necessary for economic results.

Holt's Boiler.—An ingenious form of double-ended boiler, used by Mr. Alfred Holt, consists of two oval end parts united by a cylindrical part whose axis passes through the upper focus of the oval; in the bottom of the end parts are the furnaces, which are each connected by a large tube to a combustion chamber in the middle of the cylindrical part, from which the tubes extend to the front above the furnaces, and are consequently much longer than usually found in a marine boiler. These boilers are made of great length, and practice has proved them to be very good and efficient steam generators, lasting very much longer than the ordinary double-ended boiler, being more elastic and the middle raised far above the influence of bilge-water.

Dry Combustion Chamber Boiler.—The boiler in this case is a simple cylindrical or oval shell, having the furnace extending from end to end, and the tubes over them likewise from end to end. The combustion chamber is external, and does not form an integral part of the boiler; but is built of brickwork so as to enclose one end, and form a connection between the furnaces and tubes. If two such boilers are placed back to back and some distance apart, and the space between them enclosed by brickwork, a double-ended arrangement is formed similar to the first described, except that the water and steam are not in common to the two parts, except by means of special connections. Such a boiler is very cheap to manufacture, as the wet combustion chamber with its flat sides and stays is avoided; the advantage of two boilers without their cost is obtained, and the evaporative efficiency is very high. The intense heat from the brickwork, however, is liable to crack the tube plates, unless care is taken; but if ample space is provided in the combustion chamber, there is not so much danger of this, or of damaging the tube ends.

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* The present naval practice is to have a separate chamber to each furnace in all boilers.

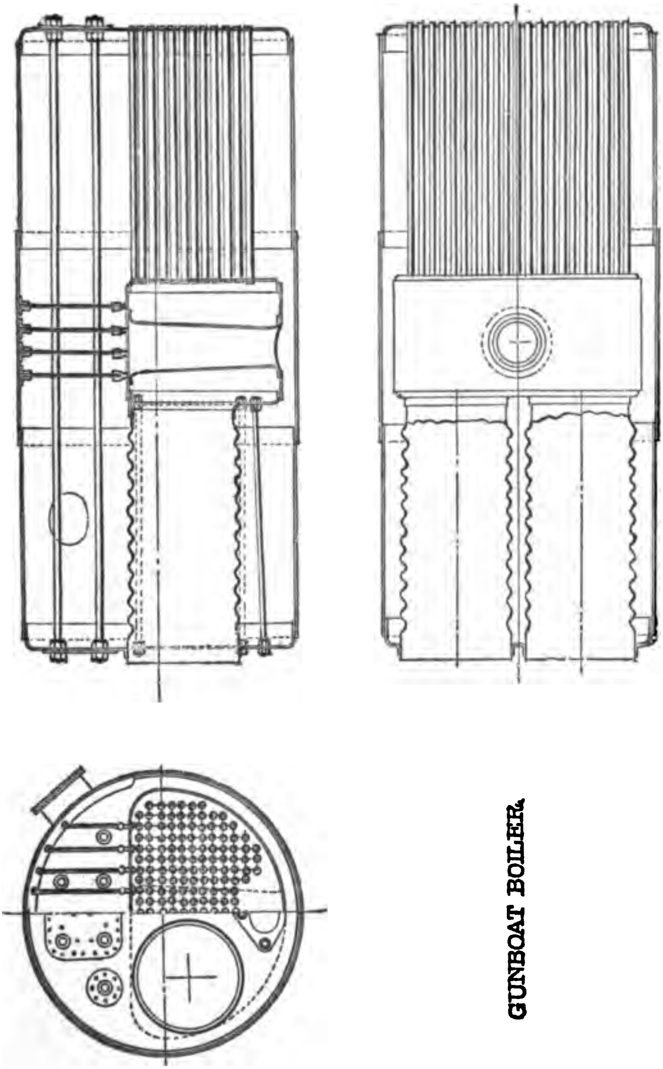

Fig. 136.—Naval Boiler, Gunboat Type.

Such boilers are now, however, not so much in use; although, among other things, they accomplished the perfect consumption of smoke, and a very even rate of evaporation. Their reputation, however, has been on

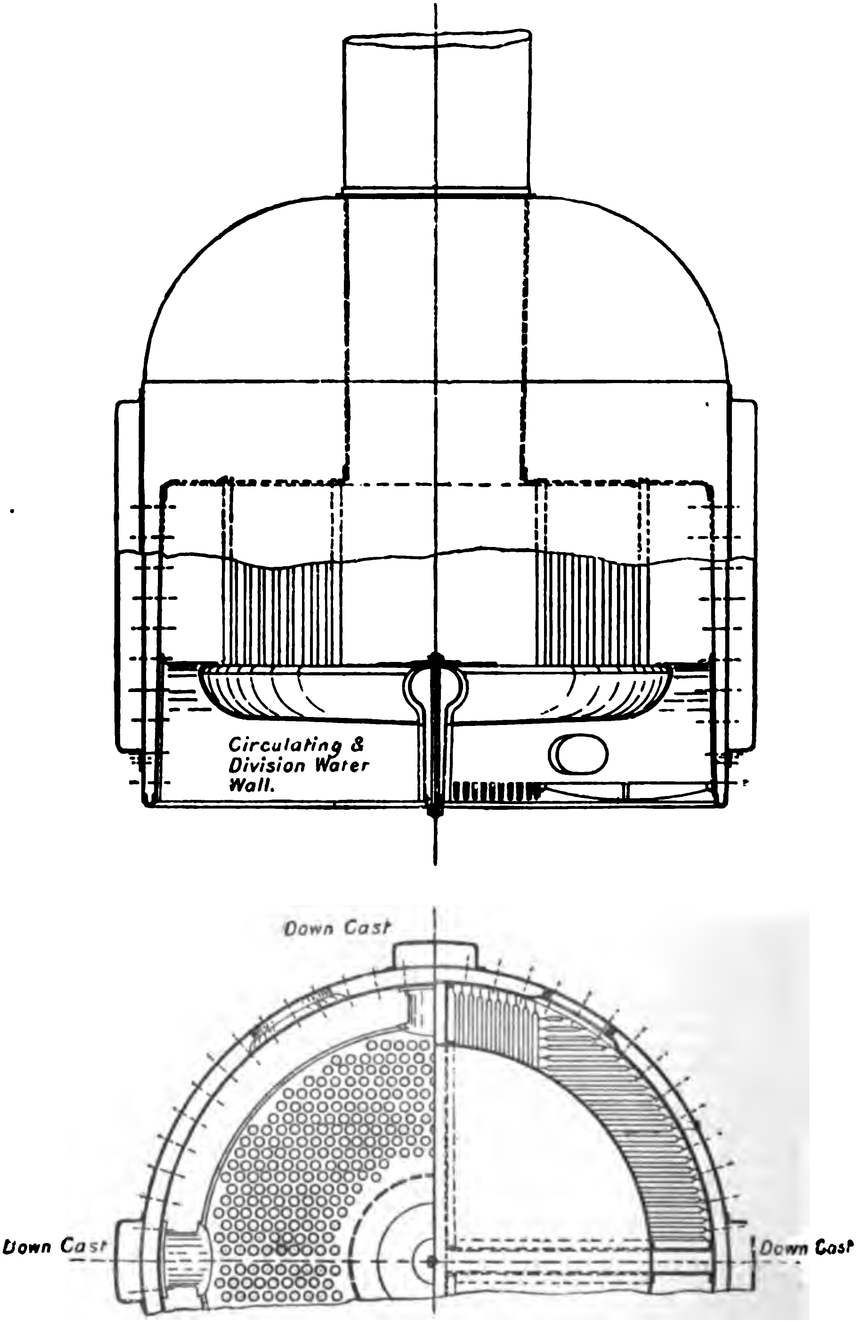

Fig. 137.—Scotch Haystack Boiler.

some occasions somewhat damaged by a too intemperate advocacy, and by improper construction and design of the combustion chamber when attempting too much with them.

For pressures above 100 pounds in ships liable to rough usage, such boilers may be adopted with advantage on account of the absence of the internal combustion chamber.

Gunboat Boilers (fig. 136).—This type of boiler, first adopted by the Admiralty for gunboats, and afterwards chosen for corvettes, is one something between the locomotive and cylindrical boiler. The shell is cylindrical, and contains the furnaces at one end and the tubes at the other, the combustion chamber being in the middle between them; the top of the furnaces is therefore level with the top of the tubes, and no part of the heating surface is far removed from the water level. The flame and hot gases flow from the furnaces into the combustion chamber, are there sometimes slightly diverted and spread by means of a hanging bridge, and flow onward with only this slight interruption into and through the tubes to the smoke-box. It is not surprising, then, that this boiler burns its coal freely, and evaporates very quickly and efficiently. Two such boilers, having a total grate area of 68 square feet, and a total heating surface of only 2200 square feet, supplied steam to triple-compound engines developing over 1340 I.H.P. The coal consumed was about 36 pounds per square foot of grate per hour; the weight of water evaporated by one pound of coal, at a somewhat reduced speed, was about 9½ pounds.

The chief objection to this class of boiler, which prevents its more general adoption in the mercantile marine, is the great length required; a stokehole is necessary at the back end to get at the tubes, and to admit of the smoke-box, &c.; and the total heating surface is also very small for the space occupied by it. It is, however, very convenient for shallow ships, when it is necessary to have clear decks, and has consequently been used for modern express river steamers.

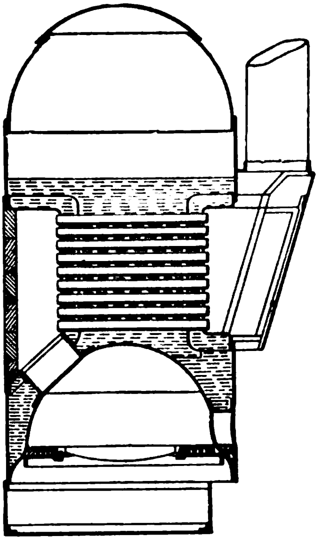
Fig. 138.—Cochran's Boiler.

Vertical Cylindrical Boiler.—This kind of boiler, with many variations of internal arrangement, is used for auxiliary purposes, and then generally called the *donkey* boiler. On a large scale it is much used by Scotch engineers for river steamers, where rapid evaporation is of more importance than economy of fuel. It is light and inexpensive in proportion to the grate and heating surface, and occupies a small amount of floor space, capacity being obtained by the height. Fig. 137 is a modern example of such a boiler as has been made for large power. Fig. 138 is a design of vertical cylindrical boiler as invented by Mr. Cochran, now of Annan, N.B., and made by him in large numbers for the auxiliary boilers of large ships and the main boiler of small ones; it is a very convenient form for these purposes.

Locomotive Boiler.—This boiler has a fire-box of rectangular section, both horizontally and vertically, enclosed in a shell of somewhat similar shape, except that the top is often semi-cylindrical. The tubes are contained in a cylindrical barrel, extending horizontally from the fire-box shell, and at the end of this is the smoke-box. As the name implies, it is similar to the boiler of the ordinary locomotive.

This form of boiler is only employed in the steam launches of the mercantile marine, but it is used in the Navy very extensively for torpedo boats as well as launches, and was even adopted on a large scale in torpedo gunboats, as well as in some of the early Destroyers. It is a very convenient form for the naval service, as it is, with the exception of water-tube boilers, the lightest kind of boiler for the heating surface contained; and as it is invariably used now with an artificial draught, the smallness of the grate area is no detriment to it. It is also especially well adapted for high pressures, as the flat surface can be stayed without affecting the accessibility, and the cylindrical barrel is of such small diameter, as to be made of very light plates for even very high pressures. Much difficulty is, however, found in keeping the tube ends tight, especially when the draught is forced much; and when pressed to their utmost capability with engines having large cylinders, they are very liable to prime excessively. The only part which presents any difficulty of construction is the furnace crown, which, being flat, requires an extensive amount of stays, &c., and as the evaporation is very rapid from it, there is great liability of a heavy scale being formed if any sea water is used, which prevents the heat from passing to the water, and causes it to destroy the plates.

It has the disadvantages of the gunboat boiler, and on that account is not likely to be used in the merchant service, except in very light-draught river steamers in the colonies, where it has for many years done good service.

Wet-Bottom Locomotive Boilers.—The success of the locomotive boiler for naval purposes caused the authorities to adopt them for a bigger class of ship, as well as for a large number of Torpedo gunboats. A modification of the design, however, was made, which converted it into a wet-bottom boiler, the air being admitted beneath the bars through an aperture in the front, as in the furnaces of the ordinary boiler. The change, however, was a most unfortunate one, as, with the exception of those fitted by Laird Brothers in H.M.S. "Rattlesnake," and others like them, this form of locomotive boiler has proved troublesome and inefficient. The circulation is bad, so that the tube-ends at the fire-box are so seriously affected as to cause considerable leaks and general trouble. The tubes are much nearer to the fire than in the ordinary locomotive boiler, and, consequently, become more readily choked; in fact, after some hours' steaming, owing to the formation of soot in the form of a bird nest at the end of each tube, the draught is so seriously impeded as to render the boiler practically useless.

The wet-bottom boiler, however, has certain advantages over the dry-bottom, and, when carefully designed, may be fairly satisfactory. Fig. 139 shows the boiler as fitted in the "Rattlesnake," whence it will be seen that the flat tubes connecting the top to the bottom of the fire-box provide a splendid means for circulation, and form a good support to the roof. The fire-bars must be lowered, at the back ends at least, so as to be in the same relative position as in the dry-bottom boiler. The fire-box itself may be divided into two independent parts by a longitudinal water-space, instead of the row of vertical tubes connecting the top and bottom, as in fig. 139. By either method an increase of heating surface is made in the fire-box, and a better means of circulation provided, as also a good supply of water to the top of the fire-box ensured.

Double-ended Locomotive Boilers.—This form of boiler consists of a fire-box with a tube-barrel at each end, which is to the ordinary locomotive boiler what the double-ended cylindrical boiler is to the single-ended. The firing-holes are on either side, and in the case of a big boiler they will be two in number, so that through the four the grate can be well worked. Moreover, from their position and size, the opening of them does not so seriously affect the tube-plates as with the ordinary locomotive boiler. This form of boiler is much lighter than the ordinary one, and is convenient in many ways, especially for vessels of limited beam.

Seaton's Patent.—By making the tube barrel of the locomotive boiler a truncated cone instead of a cylinder, and arranging the tubes so that the top rows are nearly horizontal, a better means for circulation is provided by the wider spacing of the tubes at the fire-box and the declination of the boiler barrel at the bottom, while, by the inclination of the tubes upwards, a better draught is ensured.

Dimensions of a Boiler.—The amount of grate area is the consideration which chiefly affects the choice of dimensions of boiler, and to a very large extent the number and

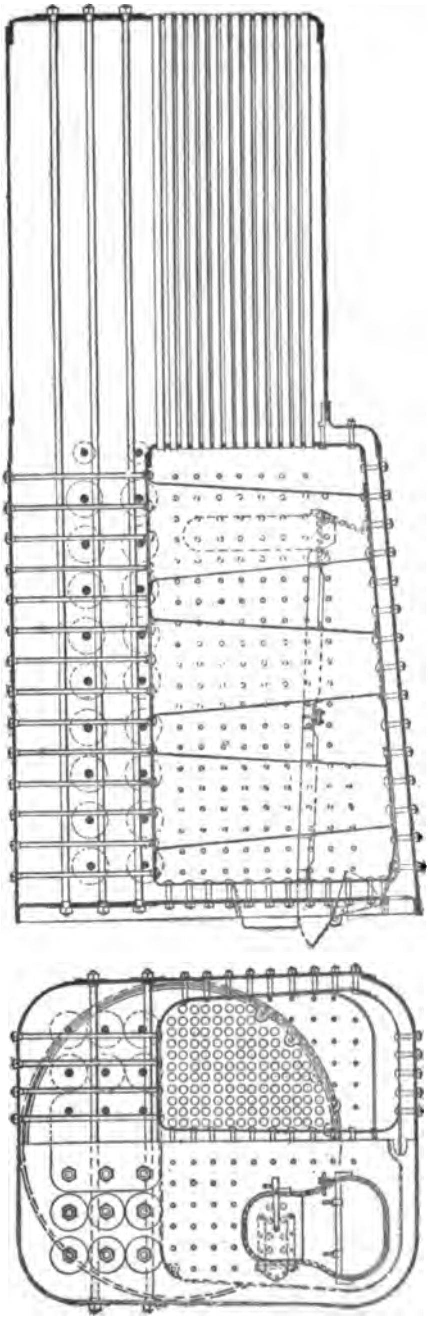

Fig. 139.—Locomotive Boiler. Marine Type (Laird's).

form of the boilers also are governed by it. The rectangular boiler could be made of any breadth without in any way affecting its length or height, so that the number of furnaces could be settled arbitrarily, and any addition to the number only meant some additional breadth. In small ships with the boiler athwartships and *fore and aft* stoking, the breadth of the ship does place a limit to the breadth of boiler, even when rectangular, but it seldom operates so as to seriously interfere with the boiler arrangement. The cylindrical boiler is not so elastic in the hands of the designer; to increase the number of furnaces in it the diameter must be increased, which means that both breadth and height are affected. If two furnaces of 40 inches diameter be the limit for a boiler of 10 feet in diameter, that there may be adequate heating surface, and 14 feet is the suitable diameter for three furnaces of 40 inches diameter, the grate bars being of the same length in both cases, the increase in boiler capacity is 96 per cent. for an increase of 50 per cent. of grate. The *smallest* diameter of shell into which three 40-inch furnaces can be fitted so as to give adequate heating surface, is 13 feet 6 inches, which is an increase in capacity of 82 per cent. over the boiler 10 feet in diameter. Four 40-inch furnaces require a shell of at least 16 feet diameter, which means an increase of 156 per cent. to obtain 100 per cent. increase of grate. To arrange four 40-inch furnaces so as to be convenient for stoking, a shell of 17 feet diameter is required, which means an increase of 189 per cent. over the shell of 10 feet diameter; if, instead of increasing the number of furnaces by increasing the diameter of shell, the number of shells be increased, the space occupied is considerably in excess of the direct ratio of grate areas.

It is true that to some extent increase of grate area may be obtained by increasing the length of furnace, but the *efficiency* of a grate in practice is nearly inversely as its length; for a long grate cannot be nearly so well attended to as a short one, nor is the air supply either under or over the bars so good with a long furnace, since the area of section at the mouth, with the same diameter of furnace, is the same whether the bars be short or long. It is no doubt for this reason that Mr. Macfarlane Gray's rule, *that the consumption of coal is very nearly proportional to the diameter of furnace*, is found to be so correct in every-day practice.

Area of Fire Grate.—The area of fire grate required for the evaporation of a certain weight of steam depends on the quantity and quality of the fuel burned on it; the quantity of coal is generally dependent to a large extent on the quality, as may be seen by reference to Table xxiv., Chapter xvii., and by the draught. It may be assumed that 1 pound of good steam coal will evaporate 10 pounds of water in the ordinary marine boiler, 7 pounds in a locomotive boiler, as fitted to torpedo boats, when not being forced, and 6 pounds when forced to the utmost; also that in the mercantile marine, where the coal is only of moderate quality, 8 to 9 pounds is a fair result, and 6 to 8 pounds only can be obtained with the coal supplied in some foreign ports. The quantity of coal burnt on a square foot of grate per hour with natural draught is about 20 pounds, under favourable circumstances; with good stoking on special bars and very good draught as much as 25 pounds may be consumed; but under ordinary circumstances and natural draught, only 15 pounds should be supplied to obtain complete combustion and economical results. With Howden's system of forced draught and warm air 50 pounds per square foot can be burnt, if the coal is fairly good and

TABLE XXV.—COMPARISON OF THE DIFFERENT TYPES OF MARINE BOILERS, CONSTRUCTED IN ACCORDANCE WITH THE BOARD OF TRADE OLD RULES FOR A WORKING PRESSURE OF 80 LBS. AND 160 LBS. PER SQUARE INCH.

General Description.	Number of Boilers.	Shell.				Furnaces.				Tubes.					Total Heating Surface.	Total Capacity.	Weight. Working Press. 80 lbs.			Weight. Working Press. 160 lbs.		
		Diam.		Length.		No.	Diam.	Length.		No.	Diam.	Length.		Surface.			Boiler.	Water.	Total.	Boiler.	Water.	Total.
		Ft.	Ins.	Ft.	Ins.		Ins.	Ft.	Ins.		Ins.	Ft.	Ins.	Sq. feet.	Sq. feet.	Cub. feet.	Tons. Steel.	Tons.	Tons.	Tons. Steel.	Tons.	Tons.
Double-ended boiler, having two combustion chambers, each common to *opposite* furnaces, -	1	12	0	14	0	4	39	5	6	376	3¼	5	6	1725	2000	1582	23·75	20	43·75	35·3	20	55·3
Double-ended boiler, having two combustion chambers, each common to furnaces at *same* end, - - -	1	12	0	14	0	4	39	5	3	376	3¼	5	3	1643	2000	1582	24	20	44	36	20	56
Double-ended boiler, having four combustion chambers, one to each furnace, - - -	1	12	0	14	0	4	39	5	3	376	3¼	5	3	1643	2000	1582	24·2	20·6	44·8	36·4	20·5	56·9
Single-ended boiler, having three combustion chambers, one to each furnace, - - -	1	14	6	9	7	3	42	6	9	274	3½	6	9	1665	2000	1582	26·15	22	48·15	37·25	22	59·25
Two single-ended boilers, each having two combustion chambers, one to each furnace, -	1	11	0	8	4	4	38	5	10	316	3½	5	10	1650	2000	1582	30·6	21·4	52	39·2	21·4	60·6

clean; with closed stokeholes, and an air pressure of ½ inch of water, 30 pounds of good Welsh and 35 of north country coal can be burnt.

From this it will be seen, (1) that the greatest weight of steam evaporated per square foot of grate per hour, under the most favourable circumstances and natural draught, is 10 × 25, or 250 pounds; (2) that with bad fuel and economical stoking it may be only 6 × 15, or 90 pounds; (3) that with fairly good fuel and favourable circumstances it may be 9 × 20, or 180 pounds, and (4) that with fairly good coal and careful stoking about 150 pounds may be expected. In practice, therefore, for short trial trips with picked Welsh coal and picked stokers, calculations may be based on an evaporation of 250 pounds; for mail steamships using good English coal, calculations should be based on an evaporation of 150 pounds for natural draught, and for forced draught from 270 to 400, depending on the air pressure; and if a ship is going to trade in the East or localities where inferior coal is to be used, the boilers should be designed on the assumption of an evaporation of only 100 pounds of water per square foot of grate natural, and 200 to 280 forced draught.

If the weight of steam required per hour for a given engine be calculated and divided by one of these numbers, the result will be the number of square feet required.

If the draught be increased by artificial means to a still higher intensity, as is the case with torpedo boats, &c., the quantity of fuel consumed per square foot of grate may be as high as 100 pounds per hour, with an air pressure of 6 inches in the stokehole; and 50 pounds with only 2 inches, the corresponding evaporations being 570 pounds and 350 pounds per square foot of grate.

Consumption of Fuel.—The consumption of fuel per I.H.P. per hour for engines working at full power was 4 pounds, with surface-condensing expansive engines, using steam of 30 pounds pressure above the atmosphere; 3¼ to 3½ pounds with similar engines of best make and large size; 2¾ pounds with compound engines when forced, and 2¼ to 2½ pounds when of moderate size and working at two-third power; 2¼ pounds with compound engines of moderate size and as generally fitted in the mercantile marine when working at full speed; 2 pounds with mercantile compound engines well designed and carefully worked at sea full speed; 1¾ pounds with large compound engines when working at sea full speed under favourable circumstances; 1½ pounds with good triple-expansion engines using English and Welsh coal of good quality, and 1⅔ pounds when ordinary steam coal is used; 1¼ to 1½ pounds with triple- and quadruple-expansion engines using steam at 200 to 180 pounds pressure; the consumption of water with these latter engines being about 12½ pounds; the consumption in torpedo boats with compound engines is 3½ to 4 pounds when working nearly full speed, and 2½ pounds with triples in Destroyers. In H.M. Navy using Welsh coal and an air pressure of ½ inch of water the consumption is 1·75 pounds including auxiliaries; with Howden's system in the mercantile marine 1·26 to 1·4 pounds of good Welsh coal with large engines and 1·33 to 1·5 with smaller is general. With super-heaters for the steam in addition to the hot air of the Howden and Ellis-Eaves systems the consumption is still lower, and generally does not exceed 1¼ pounds per I.H.P. per hour.

Assuming the consumption of coal to be 1½ pounds per I.H.P. per hour, and the grate to burn 15 pounds per square foot, there should be 0·1 square foot of grate per I.H.P. If the sea full speed I.H.P. of a merchant ship be multiplied by 0·1, the result is the grate area required for that power.

On trial trips with good coal and good stoking, and the engines working at full speed, the triple-compound engine will develop 14 I.H.P. per square foot of grate; hence, one-fourteenth of a square foot per I.H.P. may be taken as the proper allowance in designing furnaces for engines to develop a certain power on a trial trip or other favourable occasion.

As the sea full speed power is usually, on long voyages, about three-fourths that developed on a trial trip, the proportion of $\frac{4}{3} \times \frac{1}{14}$ or 0·095 of a square foot per I.H.P. developed at sea, corresponds with that given above. With the higher funnels now supplied to steamships better results are obtained, and it is found that even 0·08 square foot per I.H.P. as developed at sea is sufficient.

Heating Surface.—Strictly speaking, all surfaces exposed to heat which are capable of absorbing, and their bodies of transmitting, that heat to the water or steam are heating surfaces; but technically only certain parts are reckoned as *effective* heating surface, and the aggregate of such surfaces is called the *total heating surface.* The surface of the upper half of the furnace, or the part above the level of the fire-bars, that of the combustion chamber above the level of the bridges, and back plates, including the actual surface of the back-tube plates, are reckoned as the effective heating surface of furnaces and chambers, and are stated separately, chiefly on account of the metal forming them being three to four times the thickness of the tubes. The surface of the tubes measured externally—that is, the area obtained by multiplying the external circumference by the length *between* the tube plates, is called the *tube surface.*

The Admiralty reckon tube surface by taking the area obtained by multiplying the external circumference by the length *over* the tube plates; and in reckoning the total heating surface, the surface of the back tube plates is omitted. The calculation is in this way simplified, and the *total heating surface* is practically the same as if calculated strictly, for the surface of the parts of the tubes covered by the plates is very nearly the same as that of the back tube plates.

The *front* tube plates should be, and usually are, omitted in calculating the total heating surface, as they cannot be considered as effective.

The *amount* of total heating surface must depend on the quantity and quality of the fuel burnt on the grates in a fixed time—that is, on the quantity of heat generated in a unit of time, and also on the quality of the surface, &c. But since a grate may at some time have to burn the best of fuel, and the engine to have the advantage, the total heating surface should be adequate for such an occasion. If, however, no increase in engine power is required, the damper must be partly closed or a portion of the grate bricked up; the latter is by far the better plan on economic grounds, as the brighter the fire the more effective is the combustion, and the consumption correspondingly decreased.

Tube Surface.—When possible, there should be 1·0 square foot of copper or brass tube, and 1·33 square feet of iron tube for each pound of coal burnt per hour; that is, in the ordinary marine boiler there should be about 20 square feet of brass tubes, and about 27 square feet of iron tubes per square foot of grate.

Since on trial trip with compound engines 10 I.H.P. were usually developed per square foot of grate, there should be 2 square feet of brass, and 2·7 square feet of iron tubes per I.H.P. developed on trial trip, or 2·66 and 3·6 square feet respectively per I.H.P. developed at sea. And since with triple-expansion engines and natural draught as much as 14 I.H.P. are developed from a square foot of grate, there need only be 1·93 square feet of iron tube surface per I.H.P. in the boilers for them. When weight of boiler is of as much consideration as economy of fuel 1·75 square feet of tube surface per I.H.P. is sufficient with triple-expansion engines. With forced draught by closed stokeholes and 30 to 40 pounds of coal are consumed the heating surface per foot of grate must be accordingly increased; the tube surface should be therefore still 1·33 feet per pound of fuel, and, consequently, 40 to 53 times the grate area.

Total Heating Surface.—The ratio of tube surface to the total heating surface is about 0·8 in the single-ended, and from 0·83 to 0·88 in the double-ended cylindrical boilers; taking the average at 0·84, the following will be the allowance of total heating surface based on the above consideration:—

TABLE XXVI.—TOTAL HEATING SURFACE.

Fair Trial Trip Results.	Naval Practice.		Mercantile Practice.	
	Natural Draught.	Forced Draught.	Natural Draught.	Forced Draught.
Cylindrical boilers, total heating surface per pound of coal per hour, - - - -	1·33	1·10	1·60	1·33
Cylindrical boilers, total heating surface per square foot of grate area, - - -	27·0	30·0	33·0	40 to 60
Cylindrical boilers, total heating surface per I.H.P., ordinary compound engines, -	2·78	2·70	3·75	3·5
Cylindrical boilers, total heating surface per I.H.P., triple-compound engines, - -	2·25	1·90	2·65	2·0
Cylindrical boilers, total heating surface per I.H.P., quadruple-compound engines, -	...	...	2·42	1·95
Water-tube boilers, total heating surface per I.H.P., triple-compound engines, - -	2·40	2·30	3·75	...
Express boilers, total heating surface per I.H.P., triple, torpedo destroyers, - -	...	2·20	...	...
Cylindrical boilers, total heating surface per I.H.P., triple, long-voyage ships, - -	...	...	3·0	2·25

The total heating surface in the locomotive boiler of torpedo boats should be based on considerations similar to the foregoing; but as weight of machinery is of more consequence than economy of fuel, and as economy of fuel can be effected by working at reduced speeds, such as would be necessary from other considerations when making long runs, the heating surface is not generally so large as would be thus given. For example, a torpedo boat burns nearly 100 lbs. of coal per square foot of grate per hour when running at full speed with a *plenum* of 6 inches; by the rule given for ordinary boilers, there should be 119 square feet of heating surface per square foot of grate; in practice, however, there are only 34 square feet. When the *plenum* is only 2 inches, about 50 lbs. of coal are consumed per square foot of grate, and although the above allowance of heating surface is small for this quantity of coal burnt, it is more in accordance with what is necessary for economical evaporation, and experiments have shown that the evaporative efficiency is then nearly 20 per cent. higher than at full speed. The locomotive boiler under these circumstances is a rapid generator of steam, if not an economical one; for, with a *plenum* of 6 inches, 18 pounds of water are evaporated per square foot of heating surface, and nearly 11 pounds with the 2 inches. The modern locomotive boiler on railways has usually from 60 to 90 square feet of heating surface per square foot of grate, and the consumption of coal is about 65 pounds per square foot per hour; as much as 130 pounds of coke have been consumed on each square foot, each pound of coke evaporating 10·5 pounds of water, which is about the same quantity evaporated per pound of coal in a somewhat similar boiler. When possible, therefore, the boiler of yachts and launches, which are required to traverse considerable distances without coaling, should be designed in accordance with locomotive practice, rather than with that of torpedo boat builders. The usual allowance in torpedo boat's locomotive boilers is about 1·5 square feet of total heating surface per I.H.P. developed on trial trip; and 23 I.H.P. are developed per square foot of grate.

The following rule gives approximately the amount of total heating surface which a cylindrical boiler may have:—

$$\text{Total heating surface} = (\text{diameter of shell in feet})^2 \times (\text{length of tubes in feet} + 3).$$

Example.—What amount of heating surface can a boiler 12 feet diameter contain, the tubes being 7 feet long?

Total heating surface $= 12^2 \times (7 + 3)$, or 1440 square feet.

Example.—What heating surface should a double-ended boiler contain whose diameter is 10 feet, and the length of tubes 5·5 feet, the combustion chambers being common to opposite furnaces?

Here total heating surface $= 10^2 \times 2 \times (5{\cdot}5 + 3)$, or 1700 square feet.

If the combustion chambers are the same as in a single-ended boiler, the quantity will be the same as in *two* single-ended boilers of the same diameter and length of tubes.

Efficiency of Heating Surface.—The rate of transmission of heat from the fuel and hot gases in a boiler to the water on the other side of the metal separating them depends on the differences of temperature, the thickness of the plate, its material, but chiefly the state of the receiving and transmitting surfaces. Some few years ago elaborate and careful experiments were made by Sir John Durston and the late Mr. Blechynden independently, which demonstrated that, when the plate is clean on both sides, there is not a great difference in temperature between the sides of the plate itself, the dry side being, in practice, only 67° to 86° F. hotter. If, however, the transmitting side is dirty with scale, &c., it may be very much greater; and, if with greasy mud, it was over 500°. That a slight layer of soot or tarry matter prevented the dry side from getting very hot was demonstrated, as also that, *until the plate* is raised to a certain temperature, the flame does not come into actual contact. Further, that the *thickness* of plate when clean is not of much consequence, and is less so when dirty; it is, however, of some importance as being a regulator of the transmission of heat. Neglecting thickness altogether, it was found that the efficiency varies as the square of the differences of temperature, and the following formula may be taken as correct:—

$$Q = \frac{(T - t)^2}{a},$$

Q being the units of heat per hour per square foot of heating surface.
T the temperature of the hot gas.
t „ „ water.
a is a factor which Rankine valued at 160 to 200; Blechynden at 40 for tubes *when cleaned*, and 50 for plates to $\frac{5}{8}$ inch thick *when clean*.

There is reason to think that Blechynden is nearer the truth than Rankine for modern boilers worked on modern conditions.

There is every reason to suppose that the efficiency is seriously affected by such inequalities of surface as permit of disturbing the flow of hot gases over it and thereby bringing really more heat in contact.

Efficiency of Boilers.—Generally, it may be taken that the efficiency of a boiler is best gauged by taking the water evaporated by a pound of coal from and at 212° F. and divide it by the amount such fuel could evaporate if all its heat were used for the purpose. In practice it is, of course, impossible to utilise the whole, or even nearly the whole, heat of combustion. It is, moreover, not to be expected of certain boilers that the efficiency so measured can be very high, but it is necessary to compare them with others of their kind as well as with boilers generally.

A boiler for express service, such as that of a torpedo boat or a short-distance passenger ship, must be light, and the measure of its efficiency for this purpose will be gauged better by knowing how many pounds of steam it can produce per square foot of heating surface per hour. Now, weight is always a factor of some value on shipboard. If, therefore, the amount be multiplied by the number of pounds of water it evaporates (from and at 212° in each case), or the number of pounds of steam it produces per pound of standard fuel, the result must express the general value of the boiler for steamship purposes.

Area through Tubes.—The sectional area through the tubes, or that area through which the hot gases and smoke pass from the combustion chamber to the funnel, should not be less than one-seventh the area of grate with the natural draught, and is usually about one-fifth. Too large an area produces a slow velocity, which permits a deposit of soot and ash to form, with the consequent reduction of evaporative efficiency. Too small an area checks the draught, especially when the surfaces have become dirty. With forced draught the area through tubes can be smaller if necessary; and when it is more than one-seventh the grate (for natural draught) it is advantageous to have retarders in the tubes. These spiral strips not only prevent a too rapid flow of gas but force the hot current over every portion of the tube.

Capacity of Boiler Shell.—To contain the requisite heating surface, and to leave sufficient steam space, the boiler shell should contain 3 cubic feet per I.H.P. for the mercantile marine, and 2·5 cubic feet for the Navy and other service which has equally quick-running engines, when made for compound screw engines, and 2 and 1·6 respectively for triple engines; when for paddle engines it should be larger. In the mercantile marine a slightly larger allowance is sometimes made, especially when for steamers making long voyages. Since the *volume* of steam produced at a pressure of 210 lbs. is only about a third that at 70 lbs. when the same weight is used, the steam space for boilers working at the high pressures now obtaining may be considerably less than was usual. Good results can be got now with an allowance of 1¾ to 2 cubic feet of boiler per I.H.P. for natural draught, and 1¼ to 1½ for forced draught. In fact, the steam space must bear a relation to the high-pressure cylinder capacity in every case; the intermittent demand of the single high-pressure cylinder of a paddle engine affects the equilibrium of the boiler very differently from that of the two high-pressure cylinders of a twin screw ship running at four times the revolutions, but using the same quantity of steam.

Steam Space.—The top row of tubes in a cylindrical boiler should be not less than 0·28 of the diameter of the shell from the top. The tubes are sometimes placed higher, but there is then risk of priming from contraction of water-surface area as well as from the conformation of the sides. Priming is often due rather to contracted area of water-surface than to small steam space, although the latter is generally set down as the cause when the tubes are high. If the water-surface is so contracted that little or no part of the boiler where there may be down currents lies immediately under it, priming is sure to ensue.

The capacity of the steam space depends on the quantity of steam used in a fixed time, and on the number of periods of supply to the engines in that time. The effect of opening to the cylinders is to reduce the pressure in the steam space, and if that reduction in pressure be sensible there will be an augmentation of ebullition at that period. If the reduction in pressure is serious there will be excessive ebullition, resulting in *priming*. For this reason a slow-moving paddle engine, which takes its steam in a series of gulps, requires to have boilers with much larger steam space than is necessary for fast-running screw engines using the same weight of steam.

There should be 0·8 of a cubic foot of steam space per I.H.P. for a slow-running paddle engine, and 0·65 of a cubic foot per I.H.P. for compound screw engines, and as low as 0·55 of a cubic foot for fast-running mercantile and naval screw compound engines. Boilers for triple- and quadruple-expansion engines may have 25 per cent. less steam space than this with natural draught, and 50 per cent. less with quick-running ones and forced draught. The amount of steam space in a boiler is not dependent on the *weight* of steam used per stroke, but on the *volume*, and for that reason a boiler constructed for, say, 200 lbs. pressure does not require so much steam space as a similar boiler constructed for only 75 lbs. pressure; for if both boilers have the same grate area and heating surface, they will evaporate the same *weight* of steam, but the volumes will be as 3 to 8 nearly. If these two boilers supply steam at the same rate to engines running at the same number of revolutions, their steam spaces may be as 3 to 8 nearly.

It is no doubt on account of the high pressure and large number of revolutions of engines, together with the shaking when running, that a locomotive boiler works so well without priming, in spite of the small steam space and contracted water surface.

Area of Uptake and Funnel Sections.—Although in practice the funnel is often designed to suit the general appearance of the ship, and it is also found that, whereas some engineers prefer a small high funnel, there are others who, for good reasons, resort to the practice of making the funnel short and of large diameter; still there is undoubtedly a certain diameter and a certain height that will give the best result, and that cannot be determined from external considerations.*

* Consult *Index* for other information.

In the Navy with natural draught the funnel is usually made with a sectional area equal to *one-eighth the area of the grate* (natural draught size). In the mercantile marine a somewhat larger funnel usually obtains, the area being from one-fourth to one-sixth that of the grate; in general practice a funnel whose sectional area is one-fifth to one-sixth that of the grate, and whose top is at least 40 feet from the level of the grate, will give a very good result. The objections to a large funnel, beyond that of space occupied and cost, are resistance to the wind and large surface exposed to the cooling action of both wind and water, whereby the hot column within is partially cooled, and the draught thereby checked. On the other hand, a small funnel is liable to become excessively hot, and when the fires are freshly charged to become choked with smoke, and at all times it tends to check the draught. The funnel of a warship may be small, because it is so seldom that the boilers are urged to the utmost, and it must be as small as possible for obvious reasons. When the draught is forced either by a blast or by other artificial means, the funnel may be short, and of comparatively small diameter. The area at the base of a locomotive boiler is seldom more than one-tenth the area of fire grate, and often as small as one-twelfth. The size, for appearance sake, is now got by making the outer casing of the required form.

Area of Funnel Section should be such that (1) for natural draught Merchant service condition there are 1·25 square inches for each pound of fuel consumed per hour, or, say, 1·88 square inches per I.H.P. of trial trip.

(2) For Naval service and short Express service conditions there should be 1·7 square inches with assisted draught, and 1·25 with forced draught per I.H.P. of trial trip, or 0·9 inch per pound of coal burned per hour.

Diameter funnel in inches = $\sqrt{\text{I.H.P.} \times \text{F.}}$
For ordinary Merchant steamer F = 2·38 natural draught.
„ Naval or Express „ F = 2·16 assisted „
„ „ „ F = 1·66 forced „

TABLE XXVII.—PARTICULARS OF SOME MARINE BOILERS MADE IN ACCORDANCE WITH THE RULES OF THE BOARD OF TRADE AND LLOYD'S (PRIOR TO 1890), AND ADMIRALTY REQUIREMENTS.

Rules on which Constructed.	Shell.						Working Pressure.	Furnaces.			Number of Combustion Chambers.	Tubes.					Total Heating Surface.	Weight of			Weight per 100 feet of Total Heating Surface.			Sq. Ft. Total H.S. per Ton of Boiler.
	Material.	Diameter.		Length.		Thickness.		Number.	Diameter.	Thickness.		Number.	Diameter.	Length.		Surface.		Boiler.	Water.	Total.	Boiler.	Water.	Total.	
		Ft.	In.	Ft.	In.	In.	Lbs.		In.	In.			In.	Ft.	In.	Sq.Ft.	Sq.Ft.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	
Board of Trade, - -	Steel	16	0	11	0	7/8	80	3	48	17/32	3	250	3¼	6	6	1360	1800	32·75	22·75	55·5	1·83	1·25	3·08	54·9
Lloyd's, - - -	,,	15	6	11	0	7/8	85	4	36	17/32	2	274	3½	7	6	1858	2250	33·65	23·6	58·25	1·50	1·05	2·55	66·8
,, & Board of Trade,	Iron	14	9	11	0	1⅛	80	3	39	17/32	3	258	3½	7	0	1630	2010	33·4	21·7	55·1	1·66	1·08	2·74	60·2
,, ,, ,,	,,	14	6	10	6	1⅛	80	3	40	½	3	280	3½	7	3	1830	2175	33·05	20·0	53·05	1·52	0·92	2·44	65·9
,, - - -	Steel	14	3	10	3	¾	80	3	42	½	3	198	3½	7	0	1251	1570	25·2	18·8	44·0	1·60	1·20	2·80	62·3
Board of Trade, - -	,,	14	9	16	6	7/8	80	6	42	17/32	3	392	3¼	6	2	2021	2500	40·2	28·25	68·45	1·61	1·13	2·74	62·2
Lloyd's & Board of Trade,	,,	13	9	19	0	13/16	80	4	48	½	4	388	3½	6	10	2388	2885	40·5	32·5	73·0	1·40	1·13	2·53	71·2
,, ,, ,,	Iron	13	6	15	0	1	70	4	45	½	2	476	3¼	5	9	2286	2610	37·5	26·6	64·1	1·44	1·02	2·46	70·0
Board of Trade, - -	Steel	13	9	15	0	7/8	90	4	45	17/32	2	420	3¼	5	7½	1968	2310	34·7	26·0	60·7	1·50	1·12	2·62	66·6
,, - -	,,	12	9	15	0	1	110	4	42	½	2	380	3¼	5	10	1860	2270	35·7	23·3	59·0	1·57	1·03	2·60	63·6
,, - -	Iron	13	0	9	3	1	80	2	42	17/32	2	216	3¼	6	1	1098	1331	22·1	14·0	36·1	1·66	1·05	2·71	60·2
Lloyd's, - - -	Steel	12	0	9	6	21/32	80	2	41	½	2	172	3½	6	4	980	1205	20·6	12·6	33·2	1·66	1·00	2·66	58·5
,, & Board of Trade,	Iron	10	6	9	0	7/8	90	2	36	17/32	1	130	3¼	6	4½	687	835	15·75	9·7	25·45	1·88	1·17	3·05	53·0
,, - - - -	,,	9	6	8	6	¾	75	2	33	7/16	1	108	3	6	0	500	630	10·65	7·75	18·4	1·69	1·23	2·92	59·1
Board of Trade, - -	Steel	8	0	8	6	9/16	90	1	42	17/32	1	80	2¾	5	9	324	420	8·00	5·8	13·8	1·90	1·36	3·26	52·5
,, - -	,,	8	6	6	0	9/16	80	2	33	7/16	1	64	2½	4	0	163	250	6·25	4·7	10·95	2·50	1·88	4·38	40·0
Admiralty, - - -	,,	14	6	17	6	31/32	130	6	45	½	3	700	2¾	6	9	3335	3960	49·8	30·45	80·22	1·26	0·77	2·63	79·5
,, - - -	,,	14	6	16	6	1 1/16	130	6	41	½	3	804	2½	6	4½	3288	3868	47·6	29·62	77·22	1·23	0·76	1·99	81·3
,, - - -	,,	16	0	18	0	1 3/16	155	8	42	9/16	4	1200	2½	6	7⅝	5105	5905	70·6	43·6	114·2	1·20	0·74	1·94	83·6
,, - - -	,,	13	1	18	6	1	155	6	40	9/16	6	708	2⅝	6	9	3218	3835	46·8	26·9	73·7	1·22	0·70	1·92	82·4
,, - - -	,,	11	6	17	3	7/8	155	4	42	9/16	4	576	2½	6	4⅝	2355	2770	33·0	19·8	52·8	1·19	0·72	1·91	83·9
,, - - -	,,	10	11	17	6	29/32	155	4	36	15/32	1	524	2½	6	10½	2305	2530	29·8	16·8	46·6	1·18	0·73	1·91	84·9
,, - - -	,,	13	1	9	6	1	153	3	40	9/16	3	354	2⅝	6	9	1609	1917	26·4	13·5	39·9	1·37	0·71	2·08	72·6
,, - - -	,,	11	6	10	0	1	130	3	30	7/16	3	214	2¾	7	0½	1065	1360	21·3	12·5	33·8	1·37	0·92	2·49	63·9
,, - - -	,,	8	5	7	3½	11/16	90	2	30	½	2	98	3	5	0	385	500	8·1	5·8	13·9	1·65	1·50	3·15	61·3

TABLE XXVIIa.—PARTICULARS OF SOME MARINE BOILERS MADE (UNDER OLD RULES OF 1890).

Rules on which Constructed.	Shell.					Working Pressure.	Furnaces.			Number of Combustion Chambers.	Tubes.					Total Heating Surface.	Weight of			Weight per 100 feet of Total Heating Surface.			Sq. Ft. Total H.S. per Ton of Boiler.
	Diameter.		Length.		Thickness.		Number.	Diameter.	Thickness.		Number.	Diameter.	Length.		Surface.		Boiler.	Water.	Total.	Boiler.	Water.	Total.	
	Ft.	In.	Ft.	In.	In.	Lbs.		In.	In.			In.	Ft.	In.	Sq. Ft.	Sq Ft.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	
Board of Trade, - - -	14	9	16	6	1 11/32	160	6	42	9/16	1	336	3 1/2	6	6	3260	3750	58·2	32·0	90·0	1·55	0·85	2·4	64·4
Lloyd's and Board of Trade,	14	3	16	0	1 5/16	160	6	38 1/2	19/32	3	590	3 1/4	6	3	3074	3620	57·75	30·6	88·35	1·60	0·84	2·44	62·7
,, ,, ,,	14	0	17	0	1 1/4	160	6	40	35/64	3	466	3 1/2	6	9	2840	3375	57·10	30·0	87·10	1·69	0·88	2·57	59·1
,, ,, ,,	13	3	16	3	1 5/32	150	6	34 1/2	31/64	3	468	3 1/4	6	4 1/2	2490	3000	45·0	26·0	71·0	1·50	0·86	2·36	66·7
,, ,, ,,	13	6	16	5	1 7/32	165	6	40	9/16	3	436	3 1/4	6	4	2350	3000	51·0	26·0	77·0	1·70	0·86	2·56	58·9
,, ,, ,,	13	0	16	0	1 5/32	154	4	43	15/32	2	464	3 1/4	6	3	2417	2810	42	24·75	66·75	1·49	0·88	2·37	66·2
,, ,, ,,	12	0	16	3	1 1/8	150	4	39	9/16	2	364	3 1/2	5	5 1/2	1775	2260	38	28·2	66·2	1·68	1·24	2·92	59·5
Board of Trade, - - -	11	9	15	0	1	150	4	40	1/2	2	296	3 1/4	6	0	1740	2100	31·4	20·3	51·7	1·50	1·00	2·50	66·8
Lloyd's, - - - - -	11	2	18	3	1 1/32	150	4	38	1/2	1	208	3 1/2	7	0	1310	1685	34·3	21·2	55·5	2·03	1·30	3·33	49·1
,, and Board of Trade,	12	3	16	6	1 1/8	160	4	38	1/2	1	344	3 1/2	6	7	2033	2380	39·7	22·55	62·25	1·67	0·94	2·61	60·0
,, ,, ,,	14	6	10	6	1 5/16	160	3	42	5/8	3	232	3 1/2	8	0	1820	2250	40·0	23·0	63·0	1·77	1·02	2·80	56·3
,, ,, ,,	14	3	9	3	1 5/16	160	3	38 1/2	19/32	3	295	3 1/4	6	3	1537	1910	35·0	17·5	52·5	1·83	0·91	2·74	54·6
,, ,, ,,	14	0	10	6	1 9/32	155	3	42	9/16	3	226	3 1/2	7	3 1/2	1484	1870	37·2	19·5	56·7	1·99	1·04	3·03	50·0
Board of Trade, - - -	13	9	10	6	1 3/32	142	3	38	7/16	3	192	3 3/8	7	3	1340	1730	31·2	20·2	51·4	1·8	1·16	2·96	55·4
,, - - - -	13	0	11	0	1 9/32	150	2	48	19/32	2	176	3 3/8	7	6	1270	1590	31·3	17·2	48·5	1·96	1·03	3·05	50·1
,, and Lloyd's,	13	0	10	6	1 1/8	150	3	36	1/2	3	152	3 3/8	7	0	1024	1360	30·4	16·5	46·9	2·24	1·21	3·45	44·7
,, - - - -	12	3	9	6	1 3/8	200	2	42	21/32	1	192	3 3/8	6	6	1045	1300	27·0	13·0	40·0	2·07	1·00	3·07	48·1
,, and Lloyd's,	12	0	9	1 1/2	1 1/8	150	2	39	9/16	1	182	3 1/2	6	1 1/2	1000	1240	23·1	13·04	36·14	1·86	1·05	2·91	53·7
,, - - - -	11	9	11	0	1	150	2	46	17/32	1	128	3 3/8	8	0	990	1250	22·6	14·0	36·6	1·80	1·12	2·92	55·3
,, and Lloyd's,	11	9	9	6	1 3/32	150	2	39	9/16	2	142	3 1/2	6	6	830	1085	23·25	12·2	35·45	2·14	1·12	3·26	46·7
Lloyd's, - - -	11	6	9	6	15/16	140	2	40	1/2	2	168	3 1/4	6	6	913	1145	20·0	10·7	30·7	1·74	0·93	2·68	57·2
Board of Trade, - - -	10	6	9	3	7/8	150	2	38	1/2	2	118	3 1/4	6	3	614	810	16·2	10·0	26·2	2·0	1·23	3·23	50·0
Lloyd's, - - - -	10	6	9	3	7/8	150	2	37	5/8	1	110	3 1/2	6	0	642	800	16·2	9·0	25·2	2·02	1·12	3·15	49·4
,, - - - -	9	0	8	6	27/32	150	2	32	7/16	2	100	3 1/4	6	0	500	655	12·6	6·5	18·6	1·92	1·00	2·95	52·0
,, - - - -	9	3	9	1 1/2	15/16	155	2	31	19/32	1	104	3 1/4	6	3	541	692	13·2	8·0	21·2	1·90	1·15	3·05	52·6
Board of Trade, - - -	8	6	8	6	7/8	150	2	30	1/2	2	104	3	5	11	473	600	11·5	6·68	18·18	1·93	1·11	3·04	52·2

TABLE XXVII*b*.—LEADING PARTICULARS OF SOME BOILERS MADE SINCE 1895.

Designation.	Made in Accordance with the Rules of	Working Pressure.	Shell.					Furnaces.			Number of Combustion Chambers.	Tubes.					Total Heating Surface.	Weight of			Weight per 100 feet of Total Heating Surface.			Sq. Ft. Total H.S. per Ton of Boiler.
			Diameter.		Length.		Thickness.	Number.	Diameter.	Thickness.		Number.	Diameter.	Length.		Surface.		Boiler.	Water.	Total.	Boiler.	Water.	Total.	
		Lbs.	Ft.	In.	Ft.	In.	In.		In.	In.			In.	Ft.	In.	Sq.Ft.	Sq.Ft.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	
H D	Board of Trade,	190	16	5	17	10	1 17/32	8	40¾	19/32	...	848	2¾	7	2	4293	5112	83·0	47	130	1·62	0·92	2·54	61·6
W F	,, ,, and Lloyd's,	180	16	4	17	6	1½	8	40⅝	9/16	...	840	2¾	7	0½	4170	4572	79·5	45·5	125	1·74	0·99	2·73	57·5
P R	Board of Trade,	210	15	6	19	6	1 13/32	6	44	11/16	...	852	2⅝	7	10⅝	4540	5210	83·0	46·75	129·7	1·59	0·91	2·50	62·7
W N	,, ,,	192	15	3	18	6	1 13/32	6	43½	⅝	...	826	2⅝	7	6	4180	4839	73·75	42·75	116·5	1·52	0·88	2·40	65·6
B T	,, ,,	214	14	5	19	10	1 9/16	6	40 11/16	21/32	...	740	2½	7	11	3760	4415	75·5	41·75	117·25	1·71	0·95	2·66	58·5
C A	,, ,, and Lloyd's,	200	14	3	19	6	1 11/32	6	42	⅝	3	730	2½	7	7½	3578	4575	68·5	40·5	109·0	1·50	0·88	2·38	66·8
I D	Board of Trade,	216	14	2	19	6	1 9/16	6	40 11/16	21/32	...	612	2¾	8	0	3460	4071	72·5	40·5	113·0	1·78	1·00	2·78	56·1
C P	Lloyd's,	180	12	9	18	0	1 3/16	6	38	17/32	3	616	2¾	6	6	2826	3415	41·0	30·1	71·1	1·20	0·88	2·08	83·3
P E	Board of Trade and Lloyd's,	190	11	6	18	9	1 5/32	4	42	19/32	4	496	2¾	7	1⅝	2494	3000	43·2	22·5	65·7	1·44	0·75	2·19	69·4
P G	,, ,, ,,	180	15	0	11	9	1 11/32	3	45	19/32	3	409	2¾	8	0	2355	2860	45·0	26·0	71·0	1·57	0·91	2·48	63·5
O O	,, ,, ,,	200	14	3	11	6	1 5/16	3	44	⅝	3	238	3½	7	9	1667	2138	40·6	19·0	59·6	1·90	0·89	2·79	52·6
D N	Board of Trade,	160	14	3	11	6	1 3/16	3	44	17/32	3	240	3½	8	1½	1756	2232	37·85	19·55	57·4	1·70	0·88	2·58	56·3
Z O	,, ,, and Lloyd's,	200	13	0	11	0	1 5/16	3	38	9/16	3	234	3¼	7	6	1492	1907	34·8	17·2	52·0	1·82	0·90	2·72	54·8
S A	Lloyd's,	180	13	0	10	6	1 1/16	3	46	19/32	3	176	3½	6	9	1067	1335	26·5	16·5	43·0	1·92	1·24	3·16	50·4
D O	Board of Trade and British Corporation,	200	11	6	11	3	1⅛	2	40	19/32	2	172	3¼	7	7	1088	1327	27·3	13·8	41·1	1·91	1·04	2·95	50·0 servo tubes
L Z	Lloyd's,	200	11	0	9	6	1	2	37¾	9/16	2	132	3¼	6	3	686	880	18·85	10·5	29·35	2·14	1·19	3·33	46·7
B Y	,,	160	11	0	9	6	15/16	2	36	½	2	146	3¼	6	3	761	1000	17·6	10·2	27·8	1·76	1·02	2·78	56·8
D N	,,	200	10	0	9	6	1	2	35	⅝	2	118	3¼	6	5½	650	820	17·3	8·75	26·05	2·11	1·67	3·78	47·4
C O	,,	180	10	0	9	6	15/16	2	35	9/16	2	118	3¼	6	5½	650	820	16·4	8·75	25·15	2·00	1·67	3·67	50·0
L N	Bureau Veritas,	180	8	9	9	9	27/32	2	32	11/16	1	82	3¼	6	9	561	600	13·4	7·45	20·85	2·23	1·24	3·47	44·8

CHAPTER XIX.

WATER-TUBE BOILERS.

SUCH boilers as are composed wholly, or nearly wholly, of small tubes, are called sometimes "tubulous" boilers and sometimes "water-tube" boilers; in modern times, the latter name has been the most general. Singular to say, the same causes that have brought this type of boiler to the front in our time operated seventy years ago on the minds of the engineers of that day, so as to produce many inventions and numerous patents, some of which are the prototypes of the most successful boilers of their kind in use at the present time. Others, while displaying a very considerable knowledge of the laws governing the circulation, so necessary for the success of every boiler, and indicating a mechanical knowledge of the highest order, were almost impossible of manufacture then, and have not survived, owing to their inferiority to other designs. In the third decade of the nineteenth century, when the railway locomotive had been tried and found to be a success, many minds were engaged on designing and perfecting steam-driven road carriages; in other words, they were attempting to substitute for the stage coach and the four-horsed travelling carriage a similar conveyance with a steam engine as the motive power. Now, on an iron track the weight of the engine was not of very serious moment; but to run successfully on a turnpike road, without doing such damage as would prohibit its use, such machinery was necessarily as light as possible, because it was carried on the same four wheels as the passengers and luggage had hitherto been carried. Naturally, therefore, attention was drawn to the boiler as being by far the heaviest part of the machinery, and of that part the water was no inconsiderable item; hence, engineers availed themselves of their knowledge of the properties of the cylinder to devise boilers on the tubulous system. Of them, by far the most ingenious were those patented by the well-known Goldsworthy-Gurney in 1827; the best of which consisted of a horizontal cylindrical drum on the top, and a somewhat smaller drum at the bottom, these two being connected by down-cast pipes at the ends, and by up-cast pipes at the sides, the latter being in the shape of the letter S, and each pair together forming the figure 8 (see fig. 140). This boiler is interesting, as being the forerunner and type of the Thornycroft (fig. 149) and other similar boilers of to-day. There were other forms almost equally interesting, especially one in which the up-cast tubes connecting the upper and the lower drums being like the letter U, projecting horizontally, as in the case of the Du Temple boiler, and had the fire underneath them. In another ingenious design by the same gentleman there was a similar boiler, but with one nest of larger tubes, the bottom parts of the nest forming the fire-grate; while in yet another instance an ingenious arrangement was effected by having two sets of tubes (right and left-handed) on this system, and interlaced into one another so as to form a central furnace, the fire-bars, of course, being the lower part of the tubes. Gurney's boiler was intended for a working pressure of 130 lbs. per square inch; but even in those days it was anticipated that steam of much higher pressure might be used, and it was probably only the want of means of manufacture and the absence of a suitable design of engines for such pressures, as well as the necessity for obtaining pure water, that prevented the general use of the tubulous boiler at a very early date.

From the beginning of the nineteenth century engineers have, both in this country and America, tried the water-tube type of boiler for high-pressure steam. That made by Col. Stevens in 1805 consisted of copper tubes 1 inch diameter and about 4 feet long. From his time to the present the number of inventions of a tubulous form of boiler is almost legion, as every engineer at some time of his life has experienced the necessity to satisfy one or other of the conditions that make the tubulous boiler superior to every other form. These conditions may be laid down as—

(1) Simplicity of form of element.

(2) Simplicity of construction due to smallness and lightness of element.

(3) Great strength of element in proportion to the working pressure and consequent large margin of safety.

(4) Small quantity of water contained, especially in proportion to the water evaporated per hour.

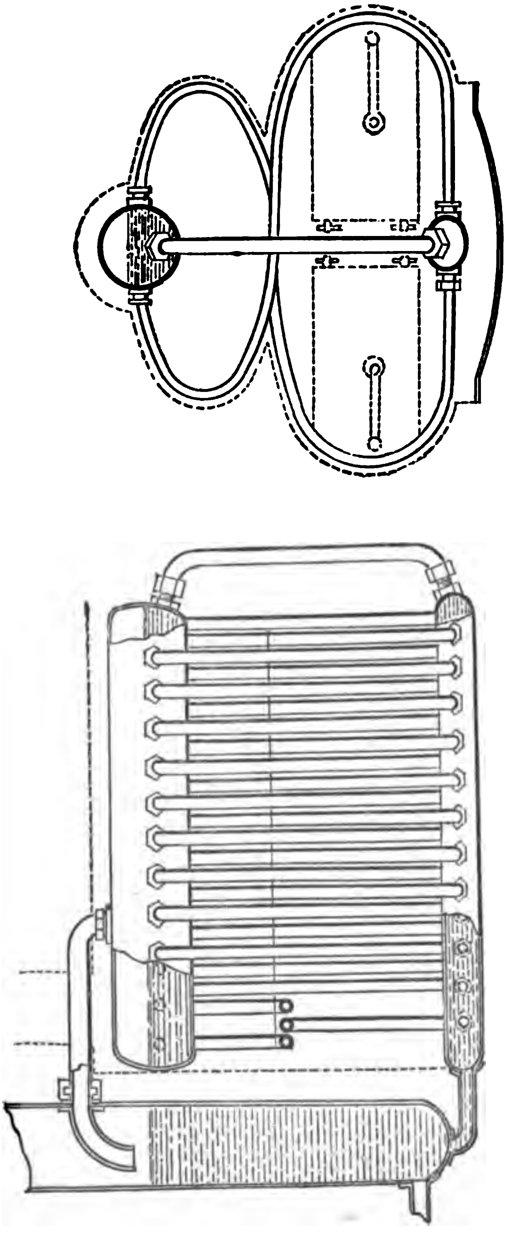

Fig. 140.—Goldsworthy-Gurney's Boiler, 1827.

(5) General lightness of structure compared with the ordinary marine boiler.

(6) Immunity from injury, and eventual destruction by (*a*) rapidity of raising steam, (*b*) forcing when at work, and (*c*) sudden cooling by drawing or putting out fires.

Every tubulous boiler worthy of notice must fulfil the above requirements; but, to be successful for everyday use, it should also satisfy the following conditions:—

(1) The elements of the boiler must be so disposed and distributed as to be capable of taking up the heat generated by the fire without allowing to pass to the chimney more than is sufficient for draught purposes.

(2) The circulation of the water must be positive, continuous, and uniform.

(3) The internal and external parts must be capable of easy examination and cleaning.

(4) Every part liable to deterioration or derangement must be capable of removal with a minimum disturbance of other parts.

(5) The steam in the receiver must be properly separated so as to be fairly dry.

The generic form of a large class of water-tube boilers is pretty much that shown on Mr. Gurney's design (fig. 140)—namely, an upper chamber, into which the steam is delivered by a series of tubes surrounding the fire, and a lower chamber or chambers connected to this upper chamber by them and by downcast tubes, the latter serving to keep up the circulation through the other tubes.

Another favourite type, however, consists of horizontal, or nearly horizontal, tubes above and around the fire connecting two chambers, on the top of one of which is the steam drum, from which a connection is made to the other, to the bottom by preference, for proper circulation.

The great attraction for this latter type of boiler is the ease with which the tubes may be examined, cleaned, or removed, and the freedom from priming when the parts are properly proportioned and arranged. It has, however, the objectionable feature of flat surfaces which, in most of the forms, require staying.

A third type consists wholly of small pipes, the steam drum (if existing at all) being in a rudimentary form. It need hardly be said that this class of boiler is the lightest, but it is likewise the most dangerous of the water tube type, and its employment is dependent on a regular supply of perfectly pure water.

The importance of the tubulous form of boiler has increased very much during the past few years, in consequence of the demand for extremely light machinery for the very high-speed vessels now required for war purposes, and the possibility of employing them successfully, is the result of the introduction of the evaporator, by which pure water can be distilled from sea water. For, although previously the surface condenser and a very small supply of fresh water carried in tanks permitted of the use of water-tube boilers in the mercantile marine, they were in almost every case a failure, owing to salt water eventually being used to make up the inevitable losses experienced in every ship; but the failure was mostly due to imperfect circulation. It is a little singular that the evaporator suggested and fitted by Hall to ships first having his surface condenser should have been allowed to go out of use and be forgotten.

The general adoption of the triple-compound engine was undoubtedly delayed for many years in consequence of the failure of the water-tube boiler in the s.s. "Propontis," and there are not wanting amongst engineers to-day those who prophesy confidently similar results for all and every water-tube boiler now in existence, or being made. But it must not be forgotten that, in past days, here and there water-tube boilers have worked successfully under the ordinary everyday circumstances of a merchant ship, including the use of salt water, and likewise that the boilers now fitted to ships have better means for examination and cleaning than existed in former times; besides which the supply of fresh water is practically ensured.

It is not claimed for these boilers that they are any more efficient as steam-generators than the ordinary marine type of boiler; but, on the other hand, it is not admitted that they are of necessity worse generators. On the contrary, it is very possible that, under certain conditions, a water-tube boiler may be more economical in steam production than the ordinary cylindrical boiler, inasmuch as it is capable of obtaining a better circulation with less energy, and is not so liable to losses by radiation. Its efficiency, however, more depends on clean surfaces and tight casings than does the tank boiler, and generally the latter is easier cleaned, &c., than it.

Hitherto, in marine practice, the heating surface in proportion to the grate has been small, and the makers and users of those boilers have been content with the other and real advantages derived from them instead of striving to effect economy by arresting the waste of heat which can, undoubtedly, take place in some forms. That the whole surface of the tubes is not efficient heating-surface goes without saying, just as no one counts as heating-surface the lower part of a furnace or flue, or the parts of a Lancashire boiler not exposed to heat. Probably only two-thirds of the circumference of the tube of any water-tube boiler acts as real heating-surface; but, for convenience of expression, the whole tube hitherto has been counted, only to serve as an argument to their detriment by hostile critics.

It must also not be overlooked that, in some designs of water-tube boilers, a considerable number of the tubes, whose surface nominally counts as heating-surface, are practically not exposed to heat at all, but really act as downcasts. Reckoning, however, the whole surface of all the tubes which are or may be exposed to heat as heating-surface, the allowance should be 40 per cent. more per I.H.P. than given in Table xxvi. for cylindrical boilers when weight is not of first consideration. In express boilers, where weight is of first consideration, as low as 1·9 square feet of nominal heating-surface per I.H.P. has given very fair results with a coal consumption of 75 lbs. per square foot of grate. It is, however, better to allow 2·2 square feet and limit the consumption to 65 or 70 lbs., then 26 to 29 I.H.P. per foot of grate is obtained, and 1·15 lbs. of Welsh coal is burned per hour for each square foot of heating-surface. That is to say, at or about 200 lbs. pressure an efficient water-tube boiler will evaporate 6 to 7 lbs. of water per square foot of nominal heating-surface per hour.

Grate Area.—The area of the grate must in this type of boiler, as in every other, depend on the draught available, and consequently the size of grate may, to some extent, follow the ordinary rules given elsewhere; but in most forms of water-tube boilers it is not possible to work economically with so high a pressure in the ashpit as is used with the locomotive or ordinary marine boiler. In some few forms, the obstruction to the passage of hot gases is sufficiently great to permit of a higher pressure, but in almost every case a large grate is necessary for the water-tube boiler, in order to distribute the heat evenly over the whole of the boiler. Moreover, it is not a good thing to have the fire too intense when any portion of the tubes is touching it or near it; in fact, the tubes which are contiguous to the fire, and receive the direct radiation from the glowing coals, should be made of somewhat thicker material as well as larger in diameter. With express boilers, and the tubes arranged nearly vertical for rapid and certain circulation, the forcing of the fires may be made by an air pressure as high as 4 inches, and 3½ to 4 inches is now the practice for full speed when the grates are getting dirty. The consumption of fuel in such cases is 65 to 70 lbs. per square foot of grate, and 2·4 to 2·6 lbs. per I.H.P. (including that required for steering gear and all auxiliaries). 26 to 29 I.H.P. per square foot of grate is obtained.

Tubes.—It is well to arrange the tubes so that access to the more remote ones permits of their receiving a fair amount of heat, and prevents those nearer the fire from getting an undue amount. As a rule, the tubes should be made of solid drawn steel, of best quality. Copper has been tried, but the difficulty of detection of a fault in solid drawn copper detracts very much from its usefulness, and is more than a set-off for the superior conductivity and the low corrosive character of that metal. The Admiralty insist on steel tubes being galvanised on the outside and not on the inside, the decomposition of the zinc when inside the tubes tending to produce hydrogen gas, by which an explosion might be caused on the incautious introduction of a lamp when empty. No doubt, for ships liable to be laying up out of use for long periods, the galvanising is an advantage; but for a ship that is generally in use, the necessity for it does not exist. There is, however, a further advantage in galvanising, inasmuch as in the process of pickling, defects can be discovered in the tubes which might otherwise escape detection. The small light boilers, employed in torpedo boats and torpedo boat destroyers, have up-cast tubes generally at least an inch external diameter and from 14 to 16 L.S.G. thick. They are expanded into the tube plates in the usual way, and when stay tubes are not used, the ends are slightly drifted so as to prevent drawing: if the tubes are galvanised, it is better to remove the zinc from the part fitting into the tube plate. For general purposes an inch tube is too small, except in steam launches; in ordinary merchant ships, as in the larger vessels for naval service, the tubes should be 1¼ ins. to 2 ins. diameter in certain forms of boilers, such as the Thornycroft, Yarrow, or Normand, and from 1½ ins. to 3 ins. in those boilers in which the tubes are horizontal, or nearly so. In the mercantile marine, where weight is not of such paramount importance, the tubes of the horizontal boiler may be even somewhat larger still, especially those in the immediate vicinity of the fire.

Circulating Tubes.—The down-cast pipes are better to be of comparatively large size and few in number, so situated as not to be exposed to heat sufficient to convert them into upcasts,* and to be in the way of the natural horizontal flow of the water current in the steam receiver. These down-cast pipes are, as a rule, 3 inches to 6 inches diameter, but when a water wall is formed by tubes placed touching one another in the main part of their length, but their ends worked zig-zag fashion at the upper and lower receivers, they may be of the same size as the ordinary tubes, these, of course, acting as down-casts.

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* Mr. Yarrow's experiments show that the down-cast pipes may be heated to advantage under certain conditions, but it is certain that the continuous flow must not be jeopardised.

Stays.—In designing a water-tube boiler the same care must be exercised, as with the ordinary boiler, in arranging the stays so as to resist deformation and destruction. Many engineers, following the example of their locomotive brethren, depend on the ordinary tubes to effect this purpose; and, no doubt, so long as they are in normal condition they may be relied upon; but when the boiler is such that the tube ends might at any time become abnormally hot and be suddenly cooled, it would not be safe to depend on ordinary tubes for holding the parts together. When straight tubes are employed, stay tubes may, of course, be fitted, as in the ordinary boiler. General stiffness, as well as local stiffness, is usually obtained by making the tube plates abnormally thick, and, when the tube plate forms the part of a cylindrical drum, this thickness is necessary for strength to resist tangential strains.

Steam Drums.—The size of the steam receiver is dependent, as usual, on the volume of steam produced, and in order to keep the size as small as possible, as well as to prevent priming, it is customary to work these boilers at a very high pressure, there being little difference in weight between such a boiler and one made for a much lower pressure. The requisite pressure at the engine is maintained by means of a reducing valve. If, however, the boiler is to be worked at the same pressure as the engines, and that pressure is only 180 lbs., the steam dome must be of considerable size; in fact, it must follow the rules laid down for ordinary boilers; it is better, however, to use water-tube boilers with higher pressures, 200 lbs. being about the lowest advisable.

The Two Types.—The modern marine water-tube boilers may be roughly divided into two classes, the one having large or comparatively large tubes and suitable for large ships and long runs, the other having small tubes and suitable for small ships or for short runs at very high speed, and consequently called "Express Boilers." Each of these can be subdivided into two divisions, the one having drowned tubes (that is, tubes whose upper ends are submerged in water), and the other having the upper ends exposed to steam only when the circulation is maintained by priming or raising of the water with the steam in the pipes, and the discharging of it into the bottom of the steam receiver. It is claimed by the makers of the latter class that the steam obtained is quite as dry as that produced by the submerged tube boiler. Of course, dash plates and other means are provided for separating the steam from the water besides the usual internal steam pipe: even with submerged tubes it is sometimes found advantageous to fit baffle plates so as to ensure steady circulation and prevent priming. The design of the priming boiler is one that admits of a greater length of tube and a longer exposure to heat than is usual with the other forms in which the tubes are straight, or nearly straight, and their advocates claim for them an elasticity and power of resisting sudden changes of temperature superior to that of the straight tube boiler. On the other hand, they have the obvious disadvantage of cost of manufacture as well as impossibility of internal examination, besides the difficulty and costliness of removing and replacing an injured tube.

Perkins' Boiler.—The Perkins family have been devoted for nearly a century to the development of the water-tube boiler, with the object of producing and using steam and water at very high temperature and pressure. The late Mr. Joseph Perkins took out numerous patents to that end, and the late Mr. Loftus Perkins, following in his footsteps, adapted these inventions to marine purposes, and eventually fitted a yacht, the "Anthracite," of considerable size, with which he successfully made a voyage across the Atlantic and back to prove the capabilities of his system. The Perkins' boiler, as generally known, consists of a series of horizontal rows of tubes, connected by vertical tubes at intervals, so as to form a complete and connected system. There is at the bottom a somewhat larger tube, into which the feed-water is pumped, and in which mud, &c., is deposited; and at the top there is also a larger tube, forming a steam receiver for the system. The fire is, of course, at the bottom, and the hot gases and flame flow among the tubes, and are spread and directed by means of baffle plates. The difficulty of circulation in this boiler is caused by the steam which is formed in the lower tubes forcing its way through the vertical tubes, and preventing a sufficient flow of water back into them; some of the steam remains in the upper part of the horizontal tubes, and, being a bad conductor of heat, does not prevent them from being burnt. The fact is, that it is essential in every water-tube boiler that the circulation be such as to rapidly carry off the steam when formed from the vessels in which it is generated, so that the surfaces exposed to fire or intense heat may be continually kept in contact with *water*.

Herreshoff Boiler.—In the very early days of the last century others, besides Mr. Jacob Perkins, attempted to produce high-pressure steam with the lightest possible apparatus, by making a boiler entirely of small tubes, and another inventor went so far as to construct one consisting of a single tube coiled roughly into the shape of a bell or beehive, and placed this over a fire. Water was pumped into the lower end of this coil, and steam emitted from the upper end. As might be anticipated, the generation of steam was most rapid, and attended with considerable risk to the onlookers. So long as the feed supply was regular and the water perfectly pure, this boiler could be used with impunity; but even now, as in those early days, no feed pump could be relied upon to work with the necessary uniformity, and as in practice, all water contains more or less solid matter, such boilers were found impracticable, and the more so as, from the curved form of the tube, it was impossible to examine or clean them mechanically. The Herreshoff boiler is a modern form of this type, which has always been a favourite one for certain purposes in America; its inventor subsequently perfected a design of it which worked with a considerable degree of success, and it was introduced by him into this country, at the time that Mr. (now Sir J. I.) Thornycroft introduced a similar boiler as regards coils, and fitted one in the s.s. "Peace" (1883). Eventually, however, it met with the fate of its prototype, so that it is now seldom or never used. His boiler consisted of two coils of pipe; the outer, which is in the form of a closed cylinder, is made of tubes of uniform section, connected at their ends by welding so as to be continuous; the inner, which is in the shape of a bell, is likewise made of continuous tubes, but those forming the sides of the bell are of larger section than that forming the crown. The top of the outer coil is connected to the top of the inner, the bottom of the outer is connected to the feed pump, and the bottom of the inner to a steam drum or receiver, from which the engine takes steam. The fire is in the inside of the bell, which is supported on brickwork, and is similar to that of a vertical donkey boiler.

The outer coil is cased in with sheet iron, and the inner is partly so covered. The water, in traversing the outer coil, is gently heated by the waste heat; on traversing the crown of the bell it is rapidly heated, so that when it enters the larger pipe it is in a state of ebullition, part steam and part water; when it has traversed the whole length of the coil it should be all steam. Another coil boiler, well-known in America, is Ward's; it has been used in the U.S. Navy, and of its kind is a good one. It, as well as Thornycroft's and Herreshoff's, has, however, the fatal fault of inaccessibility for cleaning.

Watt's Boiler.—Mr. Watt for many years gave great attention to the designing and making of water-tube boilers, and his, of their kind, have been most successful. In this, the tubes are in parallel layers, slightly inclined to the horizontal, and the ends of the whole of them connected to two narrow flat boxes. The back of these boxes is stayed to the tube plates, and opposite to each tube end is a hole through which the tube may be examined and cleaned on removing the cover. It is seen that the two boxes serve as chambers for the water and steam, as well as to connect the tubes together. The steam formed in the tubes of this boiler flows gently along their upper surface, aided by their inclined position; it finds its way into the front chamber, and rises into the upper part of it; as the steam forms and flows away from the tubes, it is replaced by water from the back chamber, into which the feed water is pumped.

D'Allest Boiler.—This boiler, which is a favourite one in France, is somewhat similar to Watt's boiler in principle and general design, though differing from it to some extent in detail.

Babcock & Wilcox Boiler (fig. 141).—The two gentlemen whose names are joined in the patent and give the title to the Company that manufactures these boilers, devoted many years to the perfecting of their system. In this case the flat boxes of the Watt boiler and of the D'Allest boiler are replaced with narrow sinuous chambers (called "headers," fig. 142), placed nearly vertically side by side. With this arrangement staying is unnecessary, but opposite the tube ends are the usual openings, as in the above-named boilers, fitted with doors or plugs. The top of each front header is connected by a pipe to a drum of considerable size, placed immediately over the back headers, and to which it is connected also by pipes. The front and back headers are connected by a large number of inclined tubes; but in the case of this boiler they are of much smaller diameter than in others of the same type. The circulation of the water is obvious and good, and of such a nature that this boiler may be classed as a priming one. The furnace is, of course, beneath the tubes, and extends nearly to the full width of the boiler. Sometimes on each side of the boiler, forming a water wall, is a system of small vertical tubes, fitted into a horizontal box at the bottom, and into a similar horizontal box of square section at the top. The top boxes are connected also to the steam drum, and at the opposite ends the top and bottom boxes are connected by external circulating tubes. These boxes and the "headers" are of square section, and made of wrought iron or steel, welded up and neatly finished. The tubes throughout are simply expanded in the tube holes, and although it appears to be a most difficult operation to thus secure them, and a still more difficult one to tighten them if they leak, in practice all such difficulties have been overcome, and these boilers found to work most satisfactorily. This, however, is now seldom supplied, as brickwork to the height of the lower tubes is better, and a good casing above in

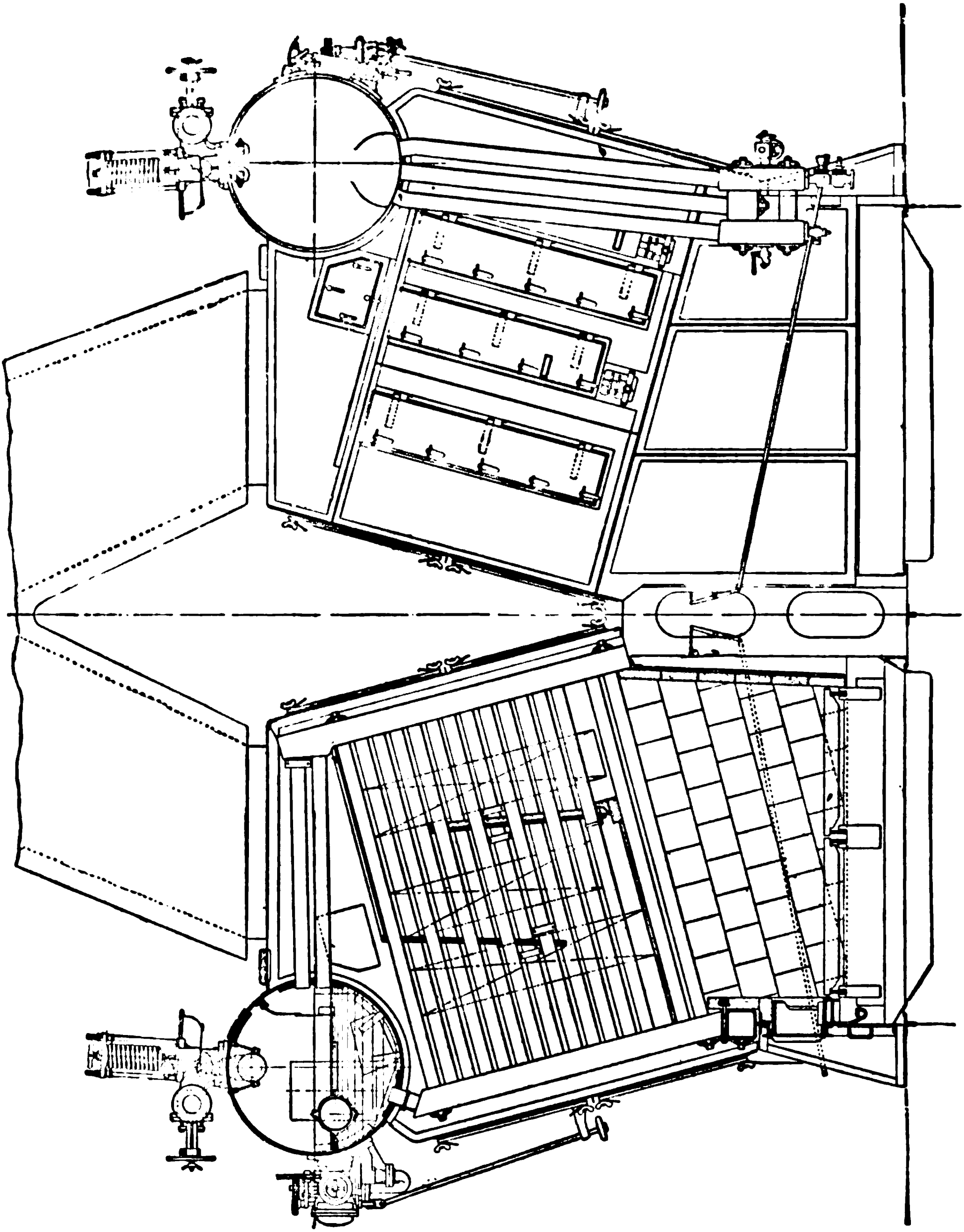

Fig. 141.—Babcock & Wilcox Naval Boiler (Large Tube).

wake of the tubes answers every purpose. It need hardly be said that the use of clean, soft water is essential. In the later marine type the tubes adjacent and near to the fires have been enlarged very considerably, so that, instead of being about 1¼ inches diameter, as were the other tubes, they are now at least 2½ inches diameter (*v.* fig. 142*a*). These boilers are lighter in

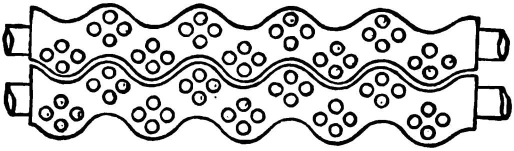

Fig. 142.—Headers for Small Tubes.

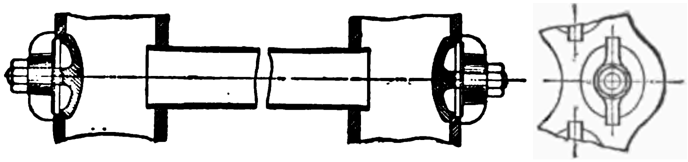

Fig. 142*a*.—Babcock Tube and Headers.

TABLE XXVIII.—BASIN AND SEA-GOING TRIALS OF H.M.S "SHELDRAKE" IN 1898-99 (BABCOCK & WILCOX).

Nature of Trial.	No. of Boilers in use.	No. of Boilers in reserve.	Duration in hours.	Steam Pressure.	I.H.P.	Air-pressure in Stokehole.	Coal per I.H.P., all purposes.	Coal per Sq. ft. Grate.	Water per Pound Coal.	Ashes per Pound Coal.	Grate Surface.
Evaporative, . . .	2	2	8	168	1116	·0	1·69	15·0	10·5	...	126
,, . . .	2	2	8	179	1292	·0	1·46	15·0	10·5	...	126
,, . . .	2	2	8	169	1761	...	1·78	25·0	9·05	...	126
,, . . .	2	2	8	165	1873	...	1·67	25·0	9·34	...	126
8 hours at 2500 I.H.P., .	4	...	8	152	2642	·0	1·43	15·0	...	...	252
3 hours at 3000 I.H.P., .	4	...	3	151	4050	·43	1·57	25·6	...	...	252
3 hours commissioning, .	4	...	3	119	2735	·14	1·64	17·8	...	...	252
1000 miles at 1500 I.H.P.,	3	1	69	120	1303	·0	1·61	12·8	...	·152	189
,, ,, 1500 ,,	3	1	68	120	1506	·0	1·6	12·67	...	·150	189
,, ,, 1500 ,,	3	1	70	135	1534	·0	1·75	14·2	...	·200	189
,, ,, 1500 ,,	3	1	68½	130	1539	·0	1·59	13·1	...	·150	189
,, ,, 1800 ,,	3	1	67	135	1829	·0	1·6	15·4	...	·216	189
,, ,, 1800 ,,	3	1	66½	140	1838	·0	1·68	16·4	...	·220	189
,, ,, 2000 ,,	3	1	59	145	2033	·0	1·57	17·0	...	·220	189
,, ,, 2000 ,,	3	1	61½	140	2042	·0	1·56	16·8	...	·230	189
,, ,, 2250 ,,	3	1	56¾	150	2245	·0	1·63	19·4	...	·250	189
Total, ...	...	...	632¼	...	...	...	...	...	...	...	...
Mean, ...	...	...	...	146	1974	...	1·63	...	9·85	·198	...

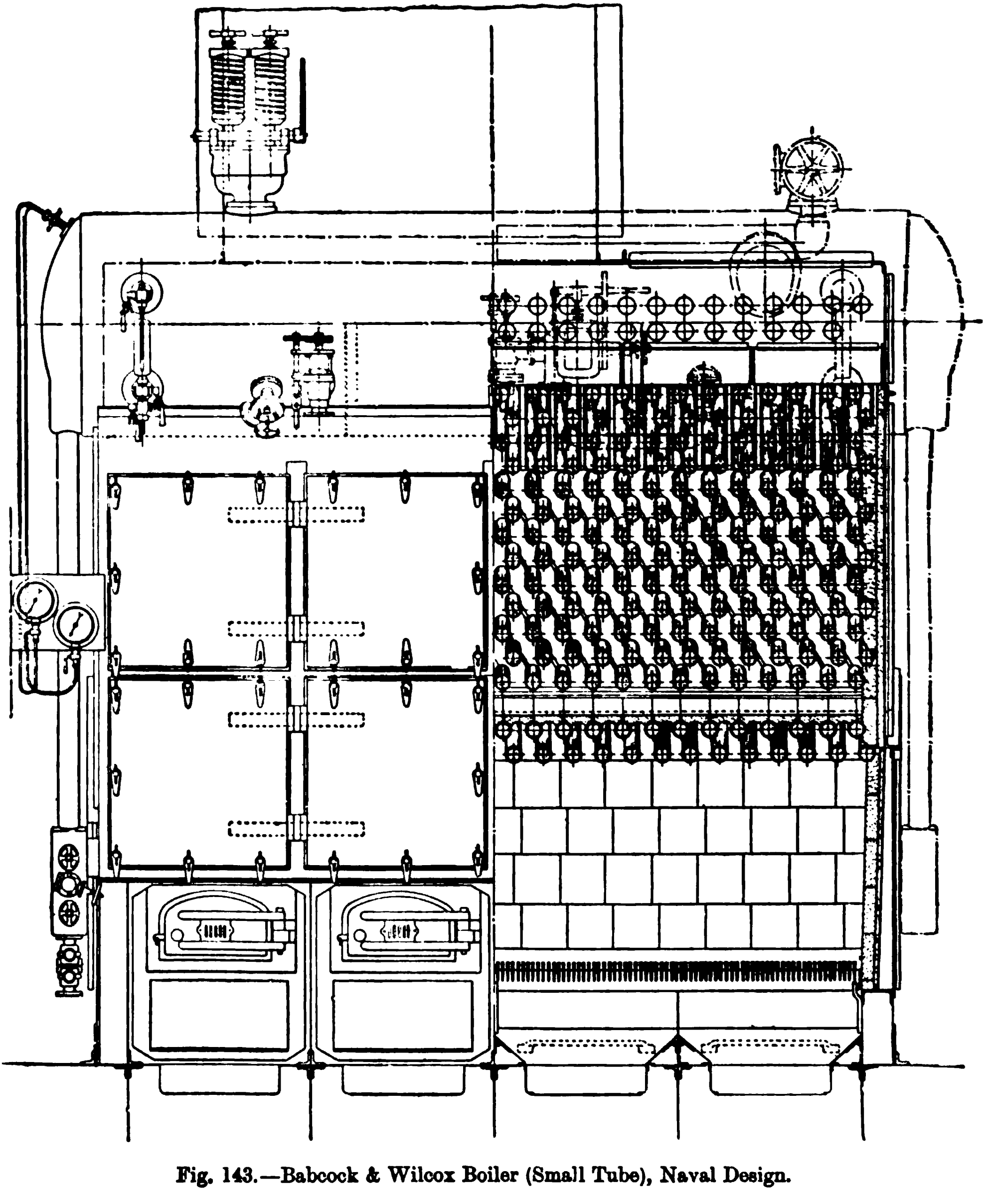

Fig. 143.—Babcock & Wilcox Boiler (Small Tube), Naval Design.

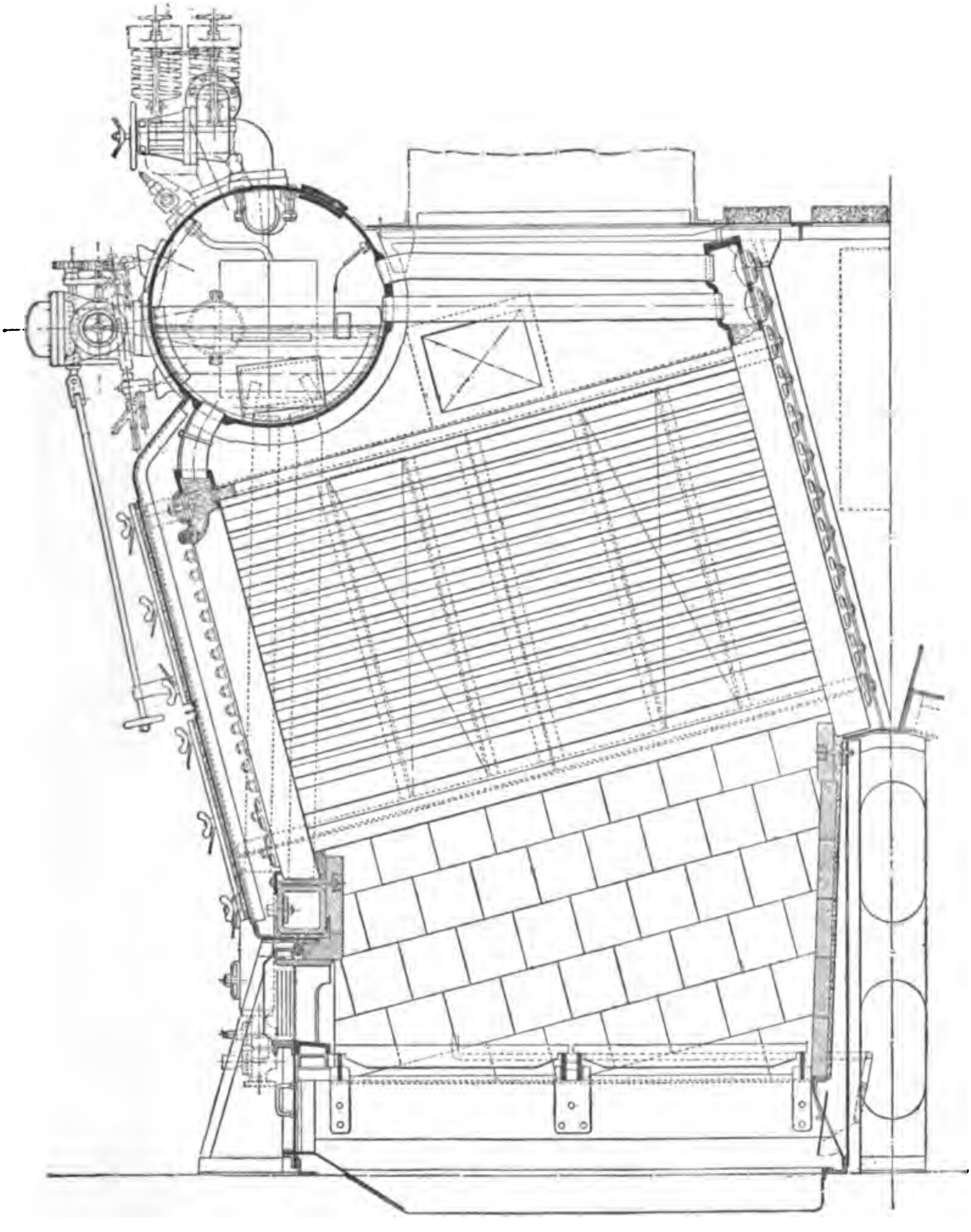

Fig. 143a.—Babcock & Wilcox Naval Boiler (Mixed Tubes)

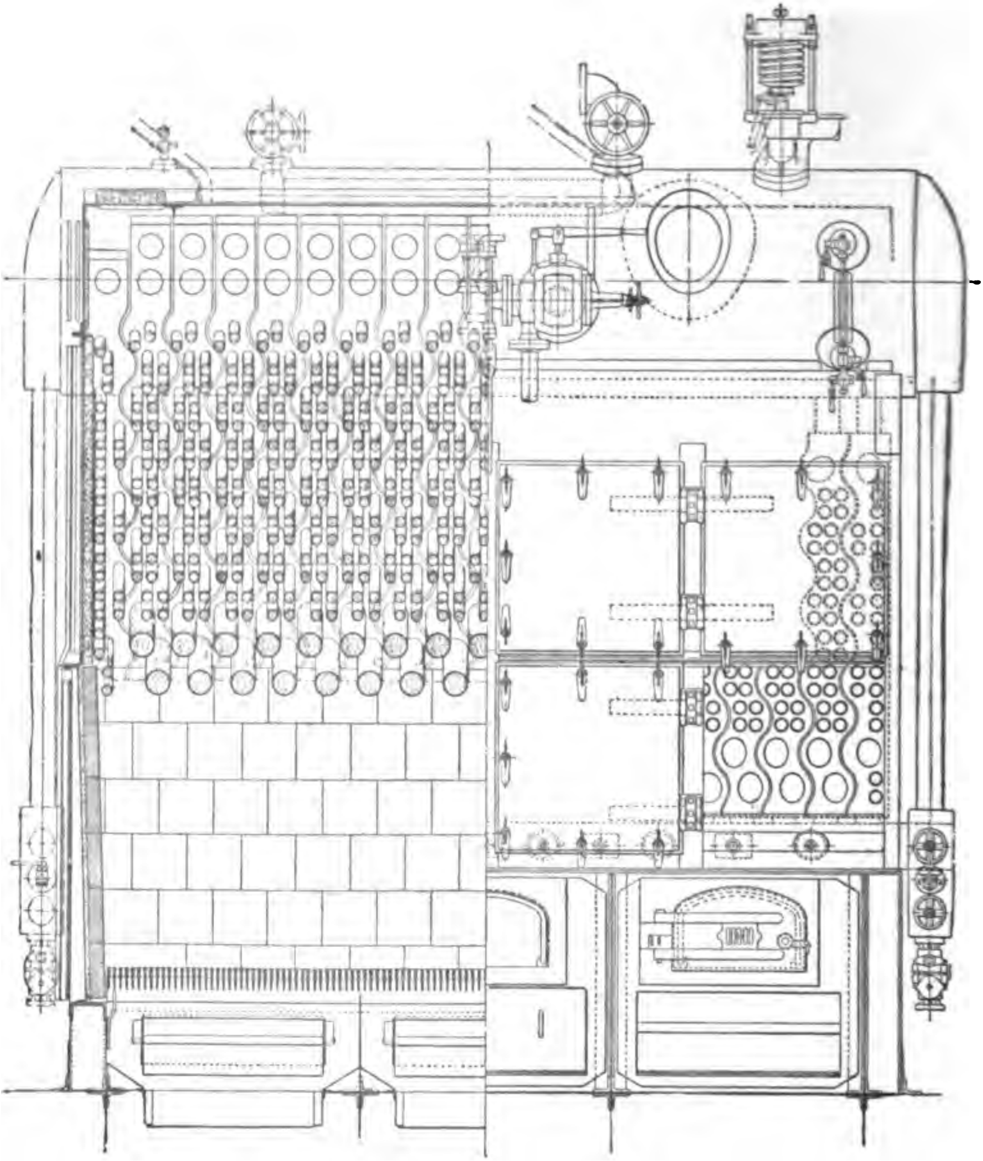

Fig. 143*b*.—Babcock & Wilcox Naval Boiler (Mixed Tubes).

proportion to their output of steam than the ordinary cylindrical boiler; but they are not so light as some of the other forms of water-tube boilers; in this case, however, there is not the same necessity as exists in others for working at a high pressure to be successful, and in this respect it is much superior to one or two of the other forms of water-tube boiler when required to be of large size. In the American Navy, as in H.M. service, these boilers have given great satisfaction, and proved themselves to be the most formidable competitors for general adoption, both against the cylindrical and the other types of large water-tube boiler. The simplicity of design and cheapness of manufacture will also recommend them for use in the mercantile marine, where, hitherto, their employment has been limited to one or two shipping firms, as soon as engineers are convinced that their length of life and efficiency of steam production is somewhat nearer to those of the cylindrical boiler than is at present believed. In H.M.S. "Sheldrake" these boilers, with best Welsh coal and a consumption of 40·7 lbs. per square foot of grate, evaporated 9·15 lbs. water per lb. of coal, from and at 212° F., and at a half that rate the evaporation was 11 lbs., the evaporation per square foot of heating surface being 8·3 and 5·0 lbs. respectively. At sea, and when running full speed, the consumption per I.H.P. was only 1·57 lbs., and at two-thirds power 1·429 lbs. Equally good results have been obtained in U.S. Naval ships with Pocahontas coal.

The Babcock & Wilcox Company sometimes fit into the uptake what is virtually an extension of the boiler, and call it a feed heater; this, of course, materially adds to the economy of the system, as it also adds considerably to the weight.

The Special Boiler Committee appointed by Parliament has reported favourably on the Babcock & Wilcox boiler, and recommended its use in the Navy; consequently a large number of ships—mostly battleships and large cruisers—are being fitted with them. The Admiralty require such boilers to evaporate 12 lbs. of water from and at 212° when the coal consumption is at the rate of 18 lbs. per square foot of grate, 11½ lbs. when the rate is 24 lbs., and 11 lbs. when 30.

Fig. 141 shows the naval boilers, which are made entirely of large tubes, and the baffling arrangements, which have been found beneficial on shore; this type is also the one best suited for the mercantile marine. Fig. 143 is a design of boiler made entirely of small tubes, suitable for cruisers and express steamers generally. Fig. 143*a* shows the boilers of the cruiser type, which are of a lighter design than fig. 141, having large tubes at the bottom only, and smaller higher up; no baffles are deemed necessary.

Normand Boiler.—Fig. 144 shows the form of boiler patented and adopted by M. Normand, of Havre. It consists essentially of one horizontal upper drum of considerable diameter, and two lower drums of much smaller diameter, the lower ones being placed on each side of the fire, and connected to the upper one by tubes of small diameter, bent to such form as to obtrude the greater part of their length into the space above the fire, and so receive heat from it, and at each end so as to be normal to the surface into which they fit. The upper and lower drums are likewise connected at their ends by larger tubes, which act as downcasts, so that the water circulates up through the smaller tubes into the upper chamber, delivers the steam with which it is charged, and flows away to the end of it, and down through the outer tubes to the water drums below. This boiler exposes a very large amount of surface, both to the direct action of the fire and to hot gases. The circulation is perfect; there are no flat surfaces requiring to be stayed; the drums themselves are of the cylindrical form, with spherical ends, so as to be as light as possible consistent with strength; and by the disposition of the tubes at the front and back there is a minimum amount of brickwork required for a boiler of this kind, while the tubes have a sufficient amount of curvature to permit of considerable elasticity in the whole structure, so as to enable it to withstand the rough usage of rapidly raising steam. On the other hand, the tubes cannot be cleaned mechanically nor examined, either internally or externally, and, in the event of a tube splitting, or becoming otherwise so damaged as to necessitate removal, unless it be an outside one, great expense is entailed by having to remove the other tubes that surround it to get at it. M. Normand arranges the inner and outer rows of tubes to form water walls on each side of the grate, and on each outside of the boiler; he leaves, however, some open work on each side of the grate near the front, through which the hot gases can pass among the tubes on each side to the back end of the boiler, where there are additional tubes massed behind the brickwork of the furnace. The hot gases then pass upward into an uptake leading to the funnel base. By this method the hot gases are made to flow uniformly over practically the whole of the tube surface. The Normand boilers in H.M.S.

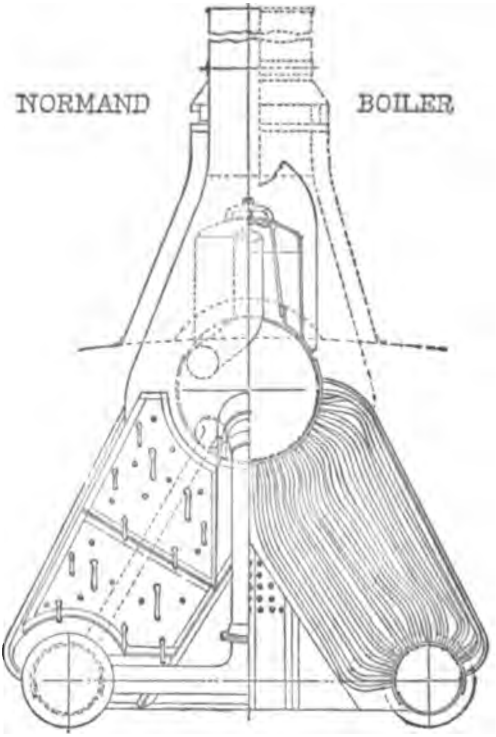

Fig. 144.—Normand Boiler.

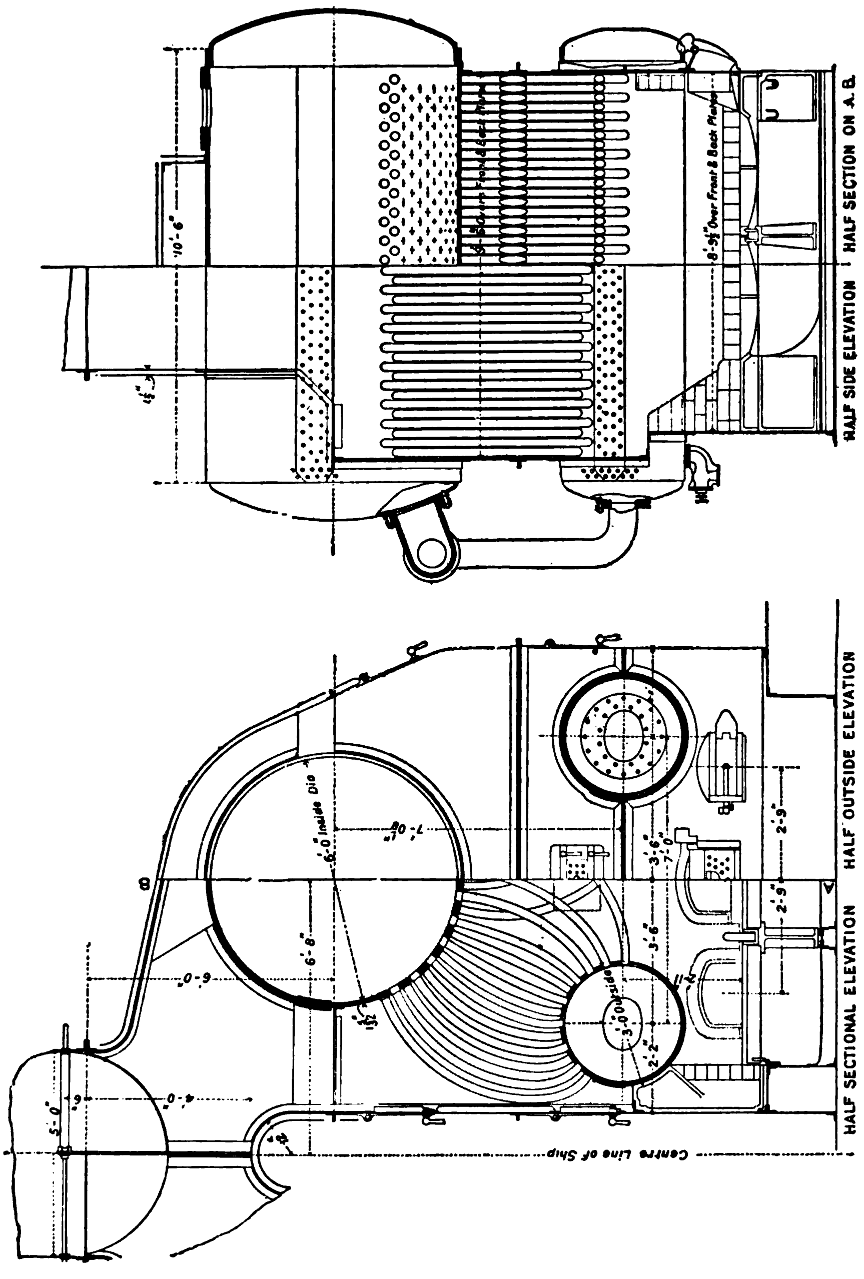

Fig. 145.—Fleming & Ferguson's Boiler (Marine Design).

"Ferret" had 8112 square feet of surface and 154 square feet of grate, and gave steam for 4774 I.H.P. Similar boilers in the French ship "Forban" proved very economical, the consumption of fuel at full speed being only 1·36 lbs. per I.H.P.

The Normand Siguady boiler consists of two Normand boilers placed back to back, and connected by the steam collectors and water drums.

Ferguson & Fleming's Boiler (fig. 145).—Messrs. Ferguson & Fleming make a boiler similar in principle to the original Normand, and get over the difficulty of drawing the tubes by limiting their length from end to end to the diameter of the upper drum, so that each and every one of them can be drawn into this drum without affecting the surrounding tubes.

Seaton's Boiler (fig. 146).—In this boiler there are two water drums at the bottom, as in the Normand, and two steam drums at the top; there is an intermediate water drum between them of sufficient size to admit of the tubes connecting it to the upper and lower drums being drawn into it and removed. In this case the tubes are straight, so that from the middle drum every tube can be examined and cleaned, as well as removed; the transverse section of the boiler is the shape of the letter X. In addition to the advantages of being able to examine, clean, or remove every tube at the middle chamber, there is in this boiler the additional one of the hot gases being forced to flow twice through the tubes before emerging into the funnel. The upper and lower drums in this boiler are connected by a water wall of tubes, which act as downcasts.

Yarrow Boiler (fig. 147).—This boiler, the invention of Mr. A. F. Yarrow, has proved to be one of the most successful in practice, both in the production of steam, and its capability of being examined, repaired, or cleaned, and it can, besides, withstand the same rough usage as the Normand. It differs from the Normand boiler, however, in having straight tubes only, so that from the upper drum they can all be examined and cleaned. In the earlier forms of it, this drum was made in halves, the top of it being connected to the lower by a flange with bolts and nuts; it was, therefore, capable of easy removal, and when this was so the bottom of the boiler consisted, as in the Normand, of two cylindrical drums; but as in the larger boilers it was found almost impossible, as well as inconvenient, to have a bolted joint in the upper drum, the lower drums were made in halves, being formed with a tube plate and pocket bolted to it, as shown in the illustration, and to remove a tube necessitates the taking off of one or other of the pockets.

The Yarrow boiler is one of the few recommended by the Boiler Committee to the Admiralty for further trials and use in large ships, but with larger tubes than usual in *Express* boilers. This kind (fig. 147*a*) has been fitted in foreign warships to be used in connection with cylindrical boilers. The lower or water drums are in this case made approximately cylindrical, like those of M. Normand, and the tubes are removed and replaced by removing those in direct line with the ones requiring it, and dodging from hole to hole as far as the slackness of the new tube in the old holes will permit.

Blechynden's Boiler (fig. 148) is like the Yarrow as now made, but the tubes are very slightly curved, and the upper end enlarged so as to permit of them being withdrawn through a few small holes on either side of the top middle line of the steam drum; in recent boilers the tubes have been arranged to draw through one set of holes on this centre line. A double

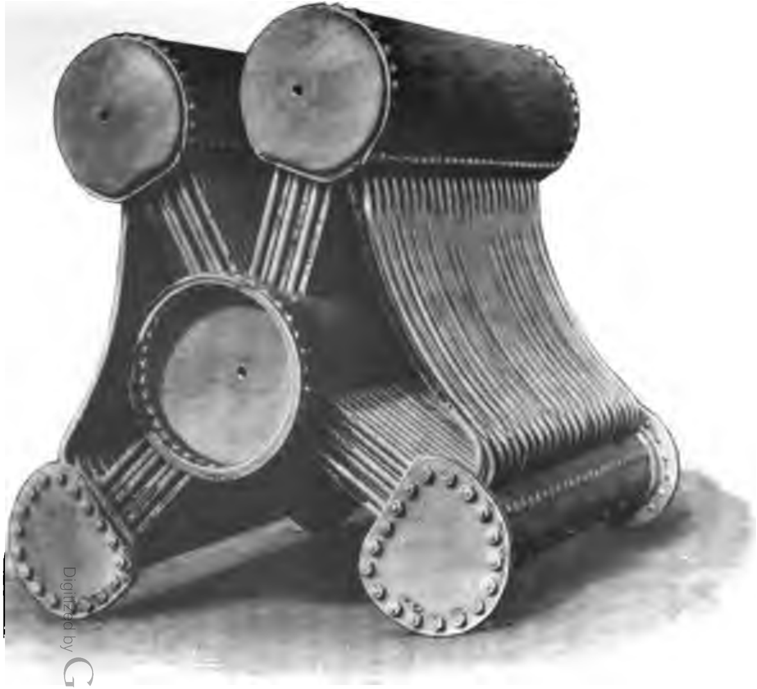

Fig. 146.—Seaton's Boiler.

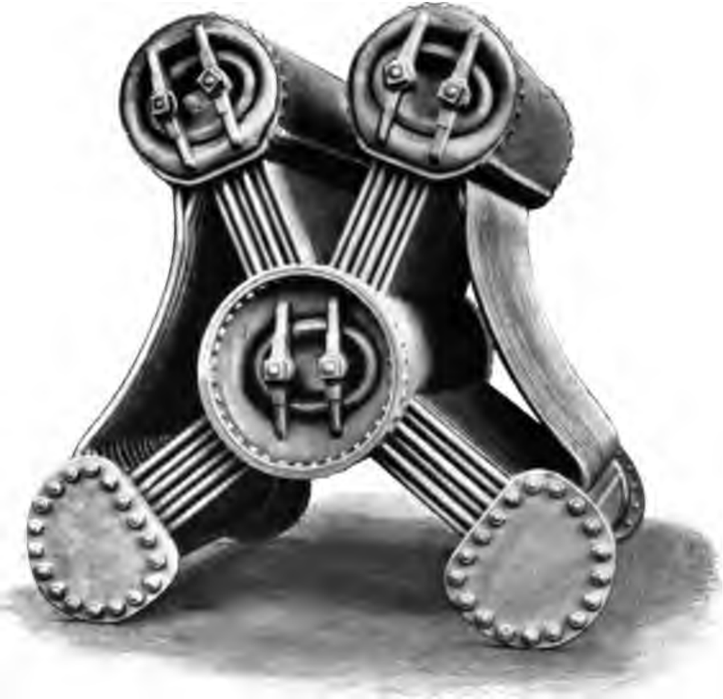

Fig. 146a.—Seaton's Boiler.

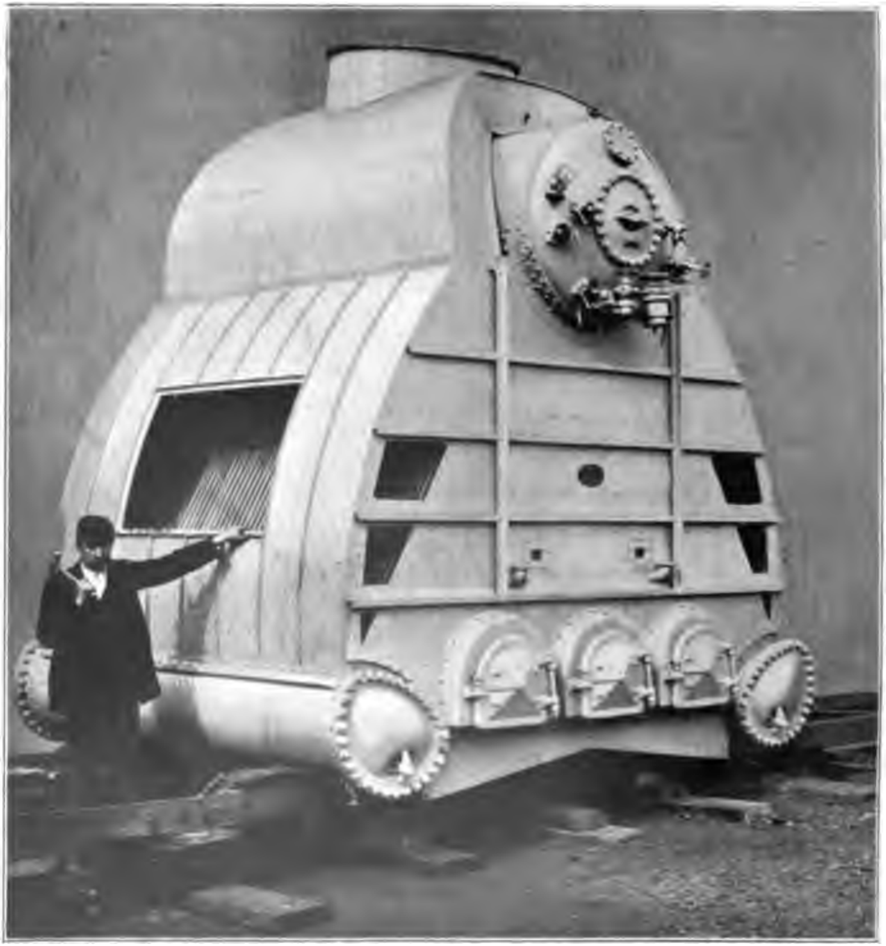

Fig. 147*a*.—Yarrow Boiler (Large Tube Type).

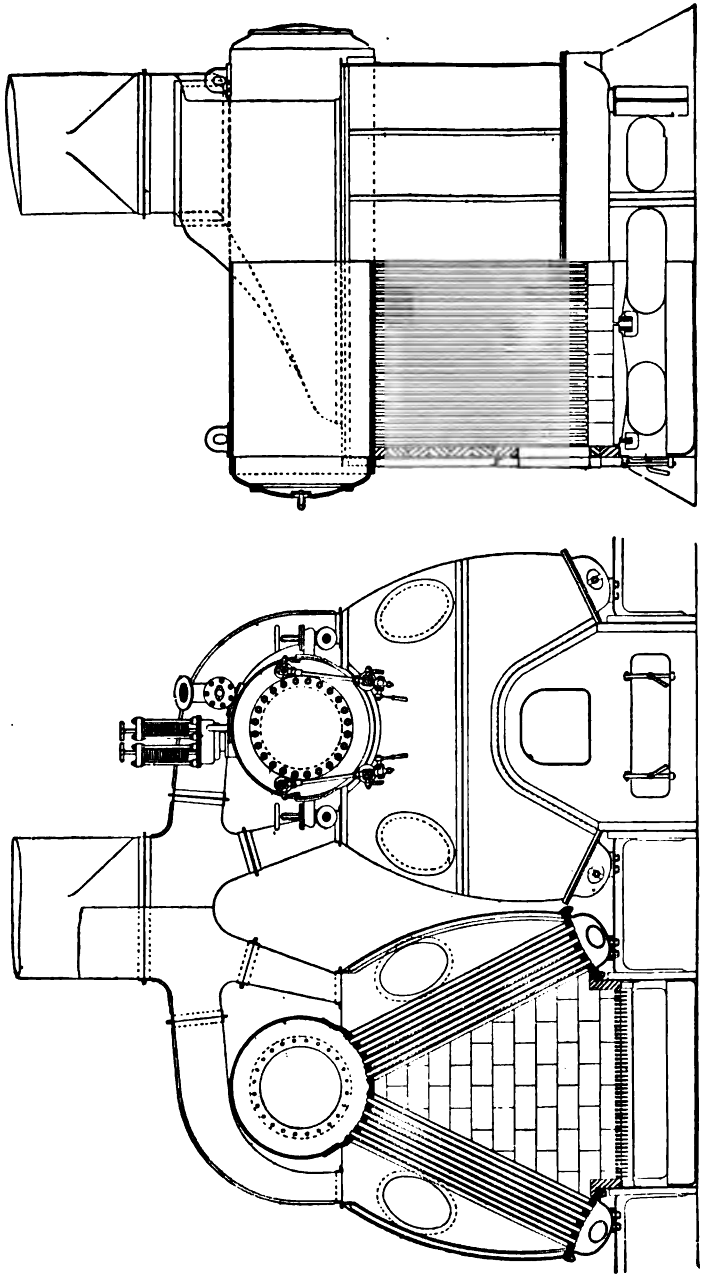

Fig. 147.—Yarrow Boiler.

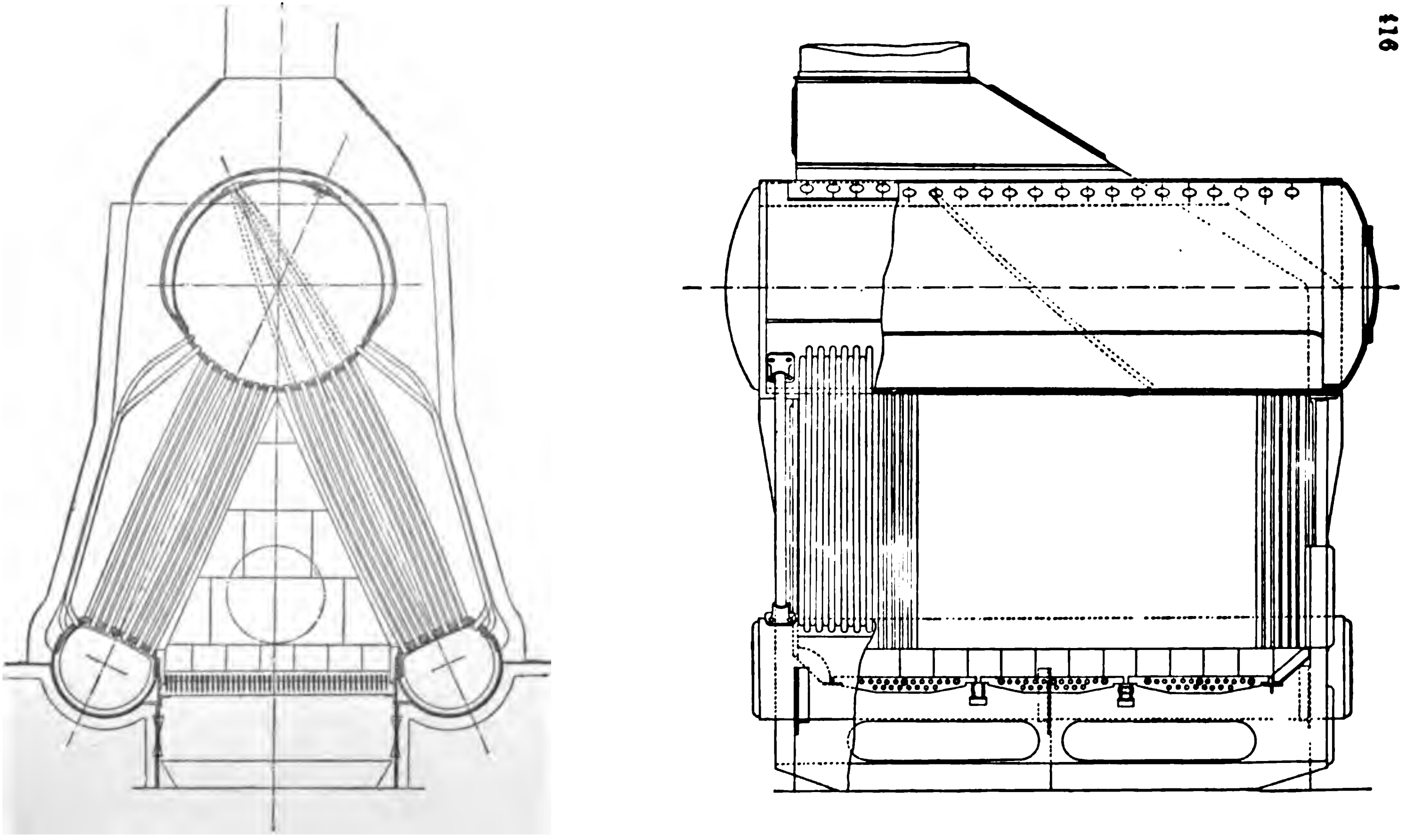

Fig. 148.—Blechynden's Express Boiler.

row of tubes forming water walls on each side, with the usual openings at the upper part, was often added to act as down-comers as well as walls.

White Forster's Boiler is somewhat like Blechynden's, but it has the tubes curved in a longitudinal plane so as to be drawn into the drum and through the manhole at its end.

Thornycroft's Boiler (fig. 149).—This boiler, known as the "Speedy" type, is the direct descendant of Goldsworthy-Gurney's invention of 1827 (see fig. 140), and consists essentially of the top and bottom drums, as in the Normand and Yarrow types, but the tubes in this case are not only bent in so as to obtrude themselves on the space over the fire, but they are carried up and around so as to enter the top of the drum instead of the bottom; it is, therefore, distinctively a priming boiler, inasmuch as there can be no circulation without the flow of water through the tubes into the steam space. There are the usual downcast pipes at each end, which are of considerable size. In this boiler there is, of course, a greater length of tube exposed to the hot gases, and consequently, if necessary, a large amount of surface per foot of grate can be provided. It has considerably more elasticity than the Normand boiler, and likewise possesses the defects of the latter, as it is absolutely impossible to examine the tubes, and, in case of damage to one of them, the difficulty of removal and replacement is considerable. Owing to the extreme curvature of the tubes, there is the greater liability of the tubes to choke with foreign matter that may be in the boiler, or get into it with the water. Experience proved it to be a very efficient evaporator, a rapid generator of steam, and capable of very rough usage. Fig. 150 is a further invention of Sir J. I. Thornycroft, and known as the "Daring" type. With the same facility for getting sufficient heating surface as with the *Speedy*, this boiler permits of a larger grate, and allows of a better circulation of hot gases as well as better uptake and funnel.

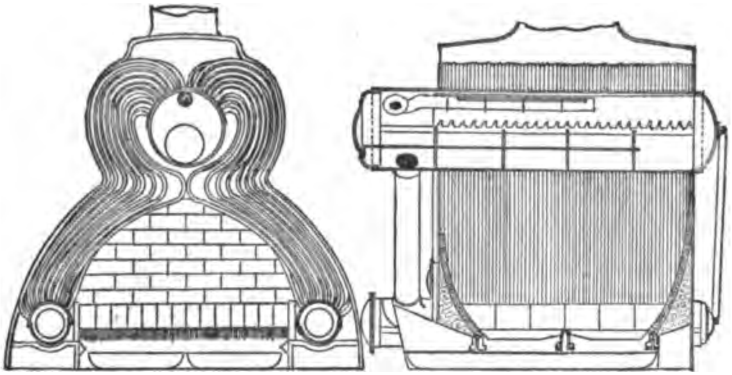

Fig. 149.—Thornycroft Boiler, "Speedy" type.

Du Temple Boiler.—This consists of the usual upper drum, which is of considerable size, and two lower chambers, of square section, placed above the fire level on each side of the grate. The top drum is connected to them by a nest of tubes bent zig-zag, so as to cross the flow of gases five times.

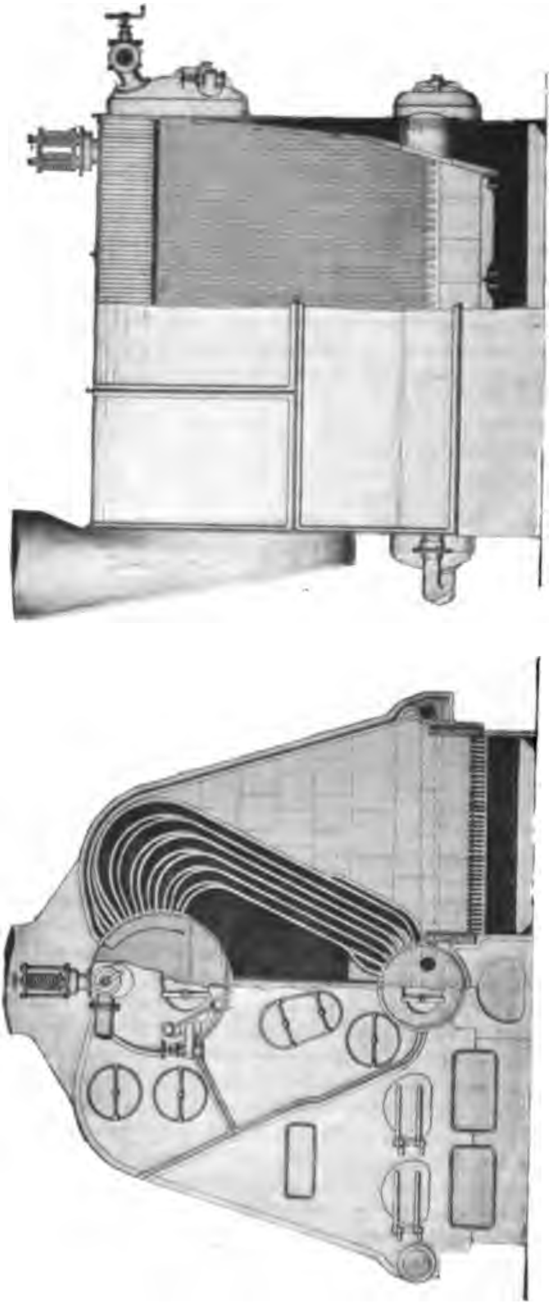

Fig. 150.—Thornycroft Boiler, the "Daring" type.

The portion of tube exposed to direct action from the fire is generally made of smaller diameter than the rest of its length. There are the usual down-cast pipes at the end. This boiler possesses many of the characteristics of the Thornycroft boiler, including some of the evils in a magnified form. The contraction of diameter of the tubes in the neighbourhood of the fire has certain advantages; but, on the other hand, modern practice has shown that there should be larger tubes exposed to direct radiation from the fire, and that those tubes in the immediate vicinity, although not exposed to direct radiation, are the better for enlargement.

Reed's Boiler.—Fig. 151 is a view of the boiler designed and patented by Mr. J. W. Reed, of Jarrow. It will be seen to consist of the usual steam receiver at top and the two wing water receivers at the bottom, both of which are in all essentials the same as found in the Normand, Thornycroft, or Yarrow boilers, but in this case they are connected differently—viz., by tubes zigzagged somewhat like

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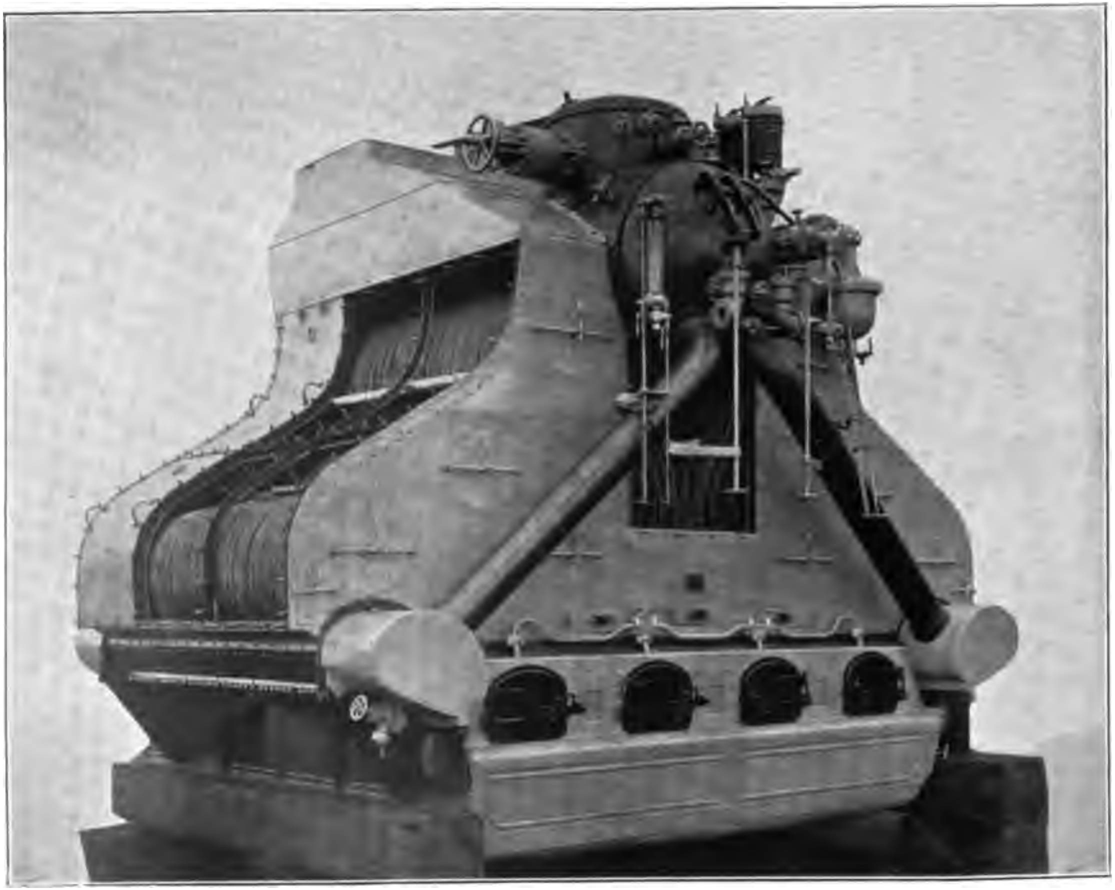


Fig. 151.—Reed's Patent Water-tube Boiler for Cruisers, Scouts, and Destroyers.

TABLE XXIX.—Express Boilers. Particulars of Surface, Weight, &c.

Name of Ship.	Type of Boiler.	Inventor.	Number of Boilers.	Total Grate Area.	Total Heating Surface.	Ratio Heating Surface to Grate.	Working Pressure.	Total Weight of Boilers, &c.	Total Weight per Sq. Ft. of Grate.	Total Weight per I.H.P.	Total I.H.P. on Trial.	Air Pressure in Stokehold.	I.H.P. per Sq. Ft. of Grate.	I.H.P. per Ton of Boilers.	Coal per I.H.P. per Hour.	Coal per Sq. Ft. of Grate per Hour.	Heating Surface per I.H.P.
				Sq. Ft.	Sq. Ft.		Lbs.	Tons.	Lbs.	Lbs.		Ins.			Lbs.	Lbs.	Sq. Ft.
H.M. Ships :—																	
"Rattlesnake,"	Loco.	Laird.	2	126	4,639	36·8	140	85·2	1515	70·0	2740	2·30	21·7	32·1	...	...	1·69
"Karakatta,"	,,	Belliss.	2	190	5,330	28·0	150	95·6	1127	55·7	3840	1·70	20·2	40·4	...	...	1·3
"Gossamer," -	,,	Sheerness.	2	183	5,654	30·8	150	103·4	1267	63·4	3654	2·70	19·9	35·4	...	...	1·5
"Speedy," -	Water tube.	Thornycroft.	8	204	17,700	86·3	...	107·2	1177	51·0	4703	1·70	23	43·9	...	...	3·7
"Havock," -	Loco.	Yarrow.	2	100	4,656	46·5	180	58·0	1299	37·0	3497	3·66	34·9	60·0	2·59	90·6	1·33
"Hornet," -	Water tube.	Yarrow.	8	154·4	8,216	53·2	180	49·0	711	28·0	3884	1·60	25·1	79·1	2·4	60·0	2·11
"Daring," -	,,	Thornycroft.	3	189	8,892	41·7	210	56·0	644	30·0	4208	3·37	22·2	75·0	3·49	77·8	2·11
"Ferret," -	,,	Laird. Normand.	4	154	8,812	52·7	175	64·2	900	32·0	4484	4·10	29·9	69·7	3·88	113	1·96
"Janus," -	,,	Reed.	4	209	10,164	48·6	210	70·6	757	41·0	3885	2·39	18·5	54·9	3·15	58·6	2·61
"Haughty," -	,,	Yarrow.	8	179	8,203	45·8	200	62·1	777	33·0	4208	2·30	23·5	67·7	2·72	63·9	1·94
"Conflict," -	,,	White.	3	212·5	11,250	52·9	185	83·0	875	38·0	4931	4·20	23·2	59·4	...	...	2·32
"Sturgeon," -	,,	Blechynden.	4	176	10,022	57·0	200	62·2	791	32·0	4367	3·1	24·8	70·1	3·29	81·6	2·29
"Quail,"- -	,,	Normand.	4	210	11,945	56·8	220	75·0	800	26·0	6347	4·25	30·2	84·6	2·48	74·9	1·88
"Desperate," -	,,	Thornycroft.	3	196	11,855	60·5	220	66·0	754	25·0	5795	3·45	29·5	87·8	2·37	70·1	2·04
"Star," - -	,,	Reed.	4	255	13,384	52·5	250	89·4	785	32·0	6318	3·32	24·7	70·6	2·53	62·7	2·11
"Gipsey," -	,,	Yarrow.	4	245	13,561	55·3	250	68	622	23·0	6523	4·4	26·6	95·9	2·51	66·8	2·07
"Avon,"- -	,,	Thornycroft.	4	236	13,400	56·8	230	71	674	26·0	6153	3·6	26·0	86·6	2·59	67·5	2·17

those in the Du Temple boiler, but not to the same extent, nor is there any reduction in the diameter of these small tubes as in the Du Temple. Mr. Reed has, besides, methods of his own for securing the small tube ends in the tube holes, whereby a tube can be easily fitted and withdrawn; and, when in place, it is capable of a certain amount of self-adjustment without any tendency to leak. This is effected by screwing the ends of the tube and fitting them with nuts formed to suit the chamfering or spherical recessing at the mouth of the tube holes.

Of the "Express" boilers none have done better than Mr. Reed's, either in respect to the quantity of steam generated per square foot of heating surface or to the number of pounds of water evaporated per pound of fuel. When severely forced, it seems to suffer little or no harm from the process, and at cruising speeds it is very economical. It, however, has the objection common to the Normand, Thornycroft, and certain other boilers, of having the tubes so considerably bent as to prevent examination internally.

White's Boiler.—Mr. White, of Cowes, employs a boiler quite distinct from any of the above, from the fact that the upper and lower drums are connected by a series of tubes bent into spirals, of about 12 inches external diameter, forming the upcasts, and exposing thereby to heat large surfaces with comparatively small height. Owing to the length of these tubes, the passage from the lower to the upper chamber will be somewhat protracted, or the speed of flow very much increased. There will, with this boiler, be a considerable amount of elasticity; but the spiral tubes are open to the same objection as the bent ones.

Belleville Boiler (fig. 152).—This boiler had been for very many years a favourite in France, and fitted into many warships, as also into a large number of their merchant ships. It had also been used extensively in America, both for marine and land purposes, when it was at last more favourably looked upon in this country, the lead in the matter being taken by the Admiralty, who replaced the wet-bottom locomotive boilers of H.M.S. "Sharpshooter" with boilers of this kind; the first-class cruisers "Powerful" and "Terrible," each of 25,000 I.H.P., and many other ships, cruisers, battleships, and gunboats have since been fitted with them. For large ships this boiler is specially suitable, and can compete successfully with the other forms of water-tube boiler in the question of weight; for smaller sizes, however, several of the before-mentioned boilers have advantages over the Belleville. It consists of the usual top drum and bottom feed collector, connected by what is virtually a set of flattened spirals; in other words, each element is of the form that a spiral spring would be if heated and flattened. There are, of course, the usual downcast pipes for circulating, besides special apparatus for preventing priming, for feeding, and for reducing the pressure. This boiler, like most of its congeners, must, to be successful, be worked with steam at a high pressure, or, to speak more correctly, with steam of low volume per lb.; consequently, the manufacturers of it usually design it for a much higher working pressure than is required, and fit it with an ingenious, and fairly reliable, reducing valve. Each element is made up of a series of tubes of considerable diameter, extending the whole length of the boiler, connected at the ends by junction boxes of malleable cast iron (fig. 152*a*), to which they are screwed, and they are zig-zagged so as to form the flattened spiral. These front junction boxes have, opposite each tube end, a hole of sufficient size to admit of the thorough examination and cleaning of the tubes, and closed

TABLE XXIX*a*.—H.M.S. "SHARPSHOOTER." SPECIAL SEA TRIALS. BELLEVILLE BOILERS.

No. of Trial.	Date.	Distance Run.	Duration of Trial.	Nature of Trial.	Indicated Horse-Power.	Speed by Log.	Number of Boilers in Use. (Total Number, 8.)	Grate Surface in Use.	Total Heating Surface in Use.	Coal.		
										Total during Trial.	Per Sq. Ft. of Grate per Hour.	Per I. H. P. per Hour.
	1895.	Miles.	Hours.					Sq. Ft.	Sq. Ft.	Tons.	Lbs.	Lbs.
1	June 20 to 22, - - -	693	48½	Sea	1553	14·3	6	202	5770	63	14·4	1·87
2	June 25 to 28, - - -	1000	69	Sea	1549	14·4	6	202	5770	91·4	14·6	1·91
3	July 3 to 6, - - -	1000	66½	Sea	1569	15·0	6	202	5770	96·5	16·0	2·07
4	July 11 to 14, - - -	1000	68	Sea	1554	14·7	6	202	5770	91·8	14·9	1·94
...	July 19 to 23, - - -	Trial abandoned on account of bad weather.										
5	July 28 to 31, - - -	1000	66¾	Sea	1804	14·9	6	202	5770	103·4	17·2	1·92
6	August 7 to 11, - -	1000	69½	Sea	1820	14·4	6	202	5770	104·2	16·6	1·84
7	Sept. 18 to 21, - -	1000	65	Sea	2016	15·3	6	202	5770	105·1	17·9	1·8
8	Sept. 28 to October 1, -	1000	63½	Sea	2037	15·7	6	202	5770	103·2	18·0	1·79
9	October 8 to 13, - -	1000	62	Sea	2194	16·1	6	202	5770	105·4	18·8	1·73

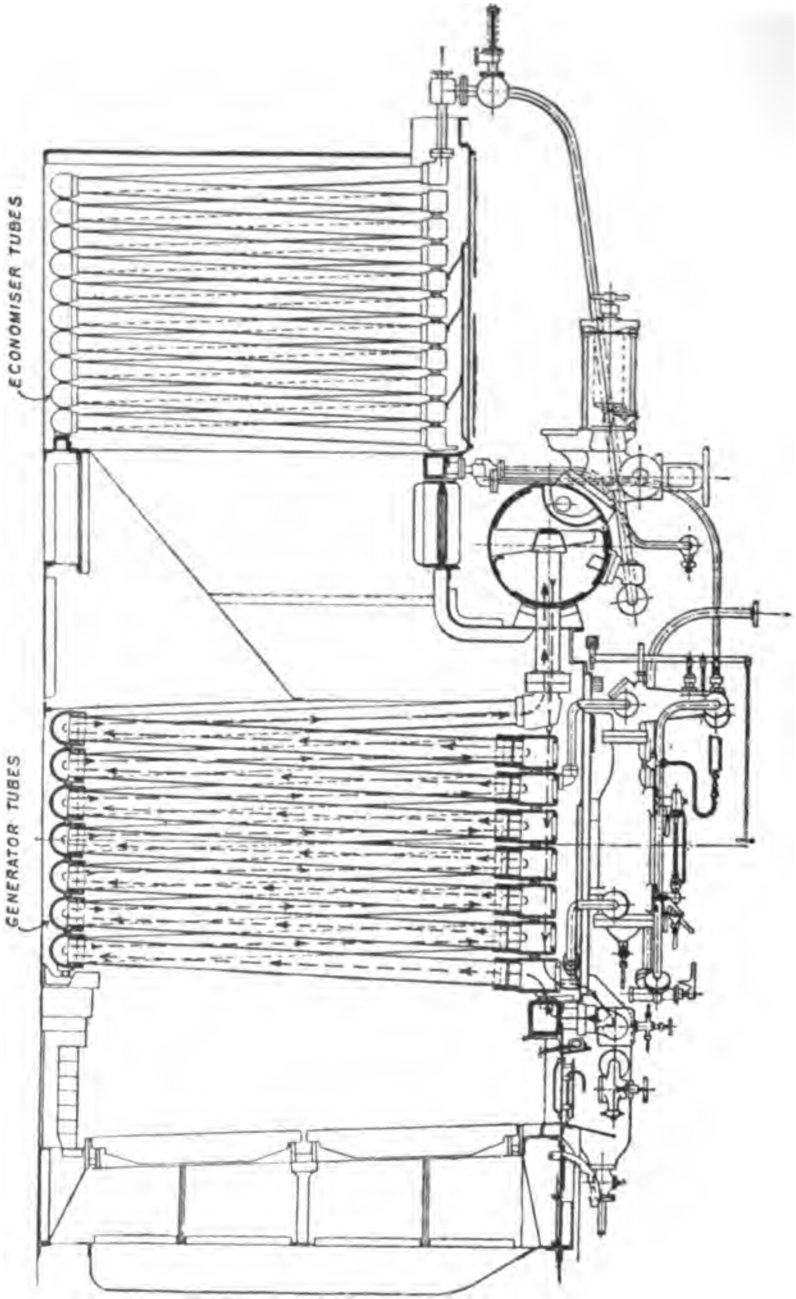


Fig. 152.—Modern Belleville Boiler, with Economiser.

with an oval door clamped in the same way as a manhole door. The top of each element enters the upper drum with a short internal pipe, and inside the drum is a series of dash plates, so arranged as to thoroughly separate the water and the steam. There is further, however, external to the boiler, a separator, having an internal spiral dash plate, and, at the bottom, an automatic drain. This boiler is a most costly one to manufacture, and even when made with an accuracy exceeding that usually followed in engine building it loses so much water by the very large number of small leaks as to seriously reduce its efficiency, as well as to require a large reserve of feed water and a constant use of the distillers. After a most lengthy and carefully made series of experiments with it and the cylindrical boiler, the Special Committee appointed by Parliament have condemned its use in H.M. Navy on these and other grounds, and at the same time, it must be said that, while recommending that all large ships should have a few cylindrical boilers for cruising and use in port, certain other forms of water-tube boilers should be further tried and worked tentatively with the object of using them in such ships solely when high speed is necessary.

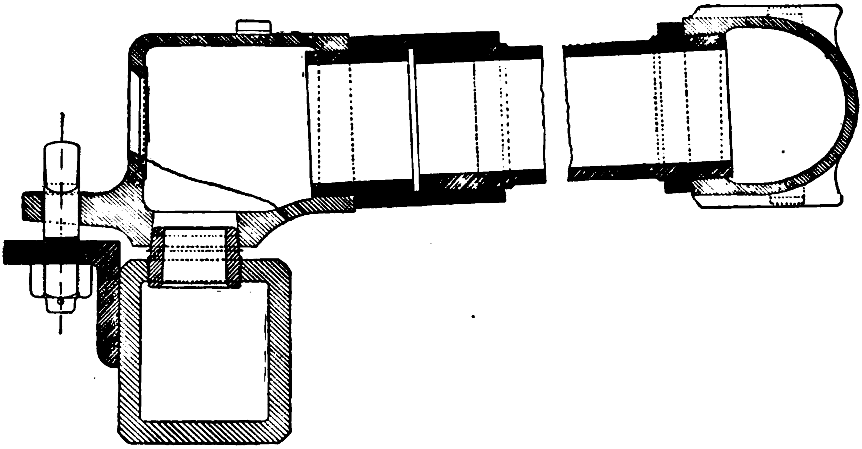


Fig. 152*a*.—Tube and Headers of a Belleville Boiler.

The boilers so recommended, besides the Babcock & Wilcox, and the Yarrow with large tubes already described, are the Dürr boiler and Niclausse boiler, both well known and used on the Continent for land purposes; the latter in the French and the former in the German Navy are favourites. Each of them has its origin in the patents of Jacob Perkins of 1831, and Howard of 1869. Perkins adopted an internal tube to cause a circulation in the large one exposed to fire; this idea was enlarged and improved by Field, by whose name such tubes are known instead of by Perkins. Howard placed his large tubes at a small angle to the horizontal and connected their front ends to a vertical chamber, and fitted them with concentric circulating tubes as far as the water level. In 1870 Miller improved on this by fitting a vertical division plate in the chamber, into which plate he fitted the circulating tubes, and placed on top of the chamber a horizontal cylindrical receiver, with which both divisions of the vertical chamber had easy communication, so that the water flowed down

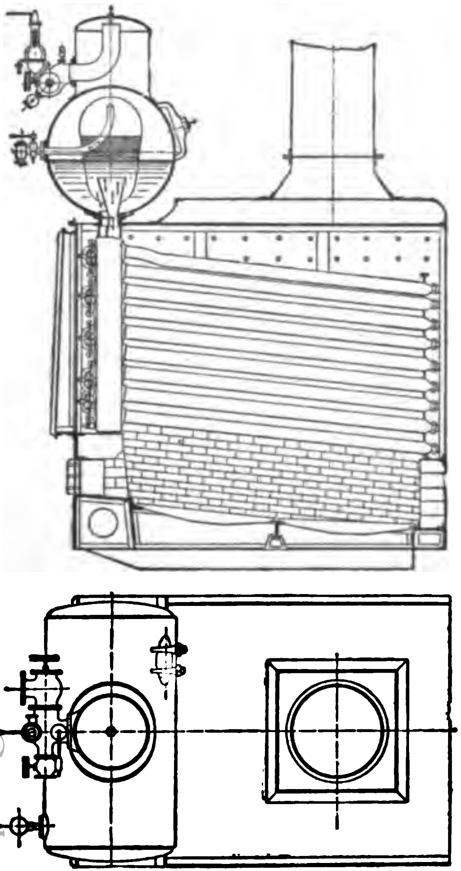
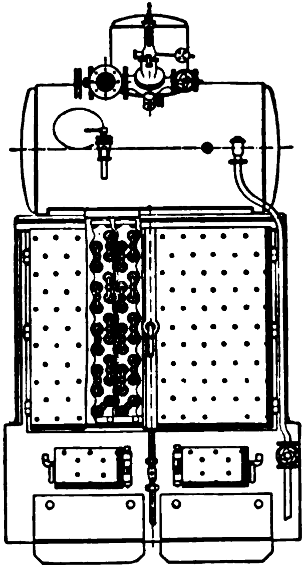


Fig. 153.—Niclausse Boiler.



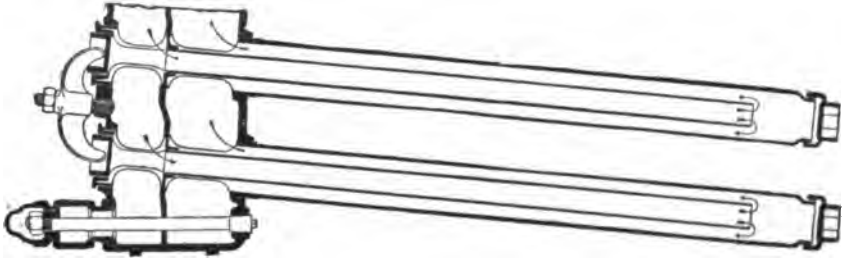


Fig. 153a.—Tubes, &c., of a Niclausse Boiler.

into the front or outer part of it and through the circulating tubes into the main tubes, and from them into the back or inner part of the chamber and thence to the receiver, where it gave up the steam, and flowed back again over the same course.

The Dürr Boiler is substantially the above, with improvements in detail.

The Niclausse Boiler differs from both Howard's, Miller's, and Dürr's in having a series of vertical boxes or headers fitted with a division plate, and having properly formed mouthpieces to allow of the tubes having a considerable slant while they themselves are vertical. These headers are made of cast steel or malleable cast iron, and connected to the under side of the steam receiver by neck pieces. The tubes in this case pass through from front to back of the headers, and fit quite steam-tight in a slightly-taper hole in the division plate and back. The portion of the tube between these is pierced with large oval holes to permit the water passing freely from it to the inner division of the header. The inner or circulating tube is fitted through a hole in the plug fitted in the end of the large tube (fig. 153*a*), and held in place by a neck piece which fits through and steam-tight in a hole in the stopper of the hole of the larger tube. The back ends of the big tubes are reduced in diameter, screwed, and fitted with cap nuts which are taken off when the boiler is to be drained dry; therein lies one of the chief objections to this and the Dürr boiler, as the time taken to remove and replace so many cap nuts is serious in all cases, and especially so in ordinary mercantile ships with a limited staff. So long as distilled water is used to make up waste there will be little need to remove them for cleaning, and it would only be in naval ships where so much harbour service would require frequent drying out.

The Stirling Boiler (fig. 154), which bears a strong family likeness to Rowan's boiler without its defects, is now coming into use for marine work after having given satisfaction to its users on shore. It has the merit of simplicity, and practically straight tubes of good size placed nearly vertically.

Three other boilers of the large tube type recently invented and patented are likely to come into general use, each possessing features showing its descent from a well-known common ancestor, and having much to commend them to the marine engineer. One of them, however, is singular in being the invention of a Japanese naval officer, Captain Miyabara, showing that that very ingenious nation is as much actively engaged in trying to solve the boiler question as are the European ones.

The Hohenstein Boiler.—This boiler (fig. 155) is a favourite one in the United States, and not unlikely to come into general use in the Navy of that country. It has been experimented upon very considerably by United States officials, and with good results. It consists essentially of two steam receivers at top, parallel with one another, and two water receivers at bottom, immediately below the steam receivers and connected to them by downcast pipes. Extending above these water receivers are rectangular section chambers or headers, and similar chambers extend from and below the steam receivers. These chambers are connected by a pair of pipes placed diagonally across the fire and connected at their remote ends by a junction box. The water, therefore, circulates by rising from the water receivers into the chambers, passing upwards through the tubes connected to it through the junction boxes, and so through the second set of tubes to the chambers underneath the steam receivers, and from them into the steam receivers themselves, and, finally, back again by the downcasts to the water receivers. It will be seen that the circulation is positive and regular The tubes thoroughly intercept the heat in its passage from the grate to the chimney. They are straight, and easily examined, cleaned, or removed.

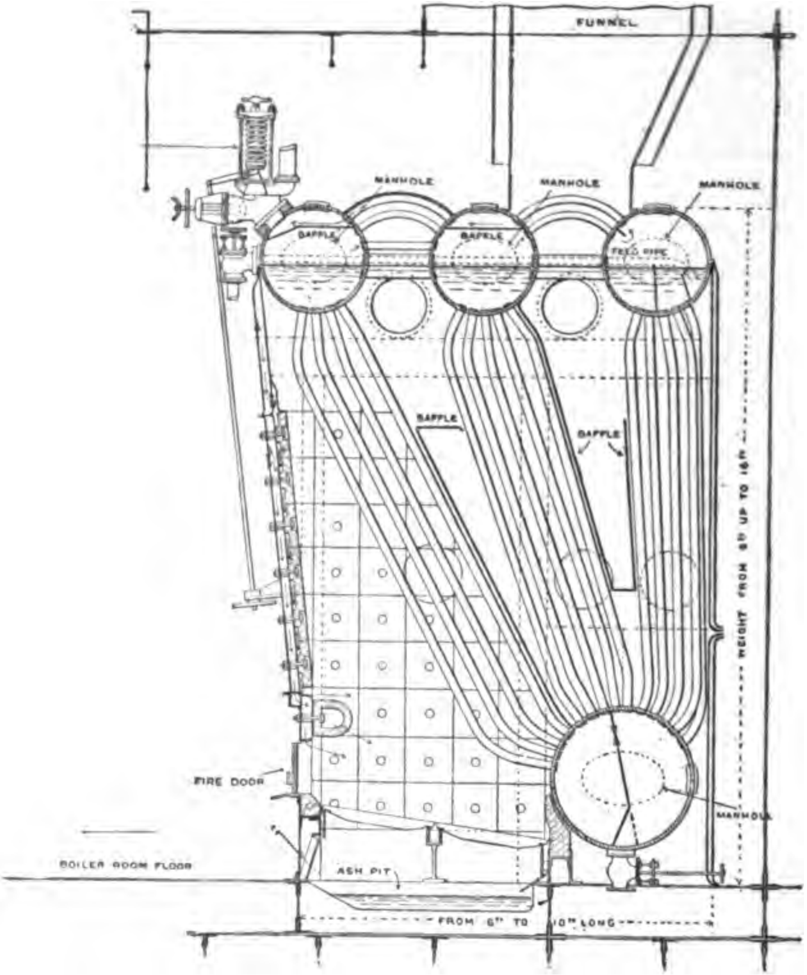


Fig. 154.—Stirling Boiler (Marine design).

The Miyabara Boiler is somewhat like the Hohenstein, but consists of two sets of cylindrical drums placed parallel, and some 6 to 8 feet apart; each set consists of 3, one above the other. The middle drum of each is connected to the top and bottom drums of the opposite set by tubes 2 inches to 3 inches diameter. The top drum in each set is connected to the bottom one by downtakes. It is very simple, and easy to make, having straight tubes with a good inclination; the hot gases will be well broken up, and distributed, and limited spaces are provided for them to come in contact with the outer casing.

The Thornycroft-Marshall Boiler (fig. 156) is the third new one, with large tubes, that seems a good design for marine or land. Its sectional form, which is the better for ship use, consists of a steam drum, with headers standing vertically downwards, capable of taking two vertical rows of tubes, and spaced so that similar headers standing upwards from the feed collectors or water drums may fit between them. The bottom hole in the lower header is fitted with a tube, whose back end fits into a junction box like that of a Belleville boiler; another tube is fitted into this box, and the bottom hole of the header from the steam drum, and so makes the connection from the feed to the steam receiver; and so on with the other elements, as

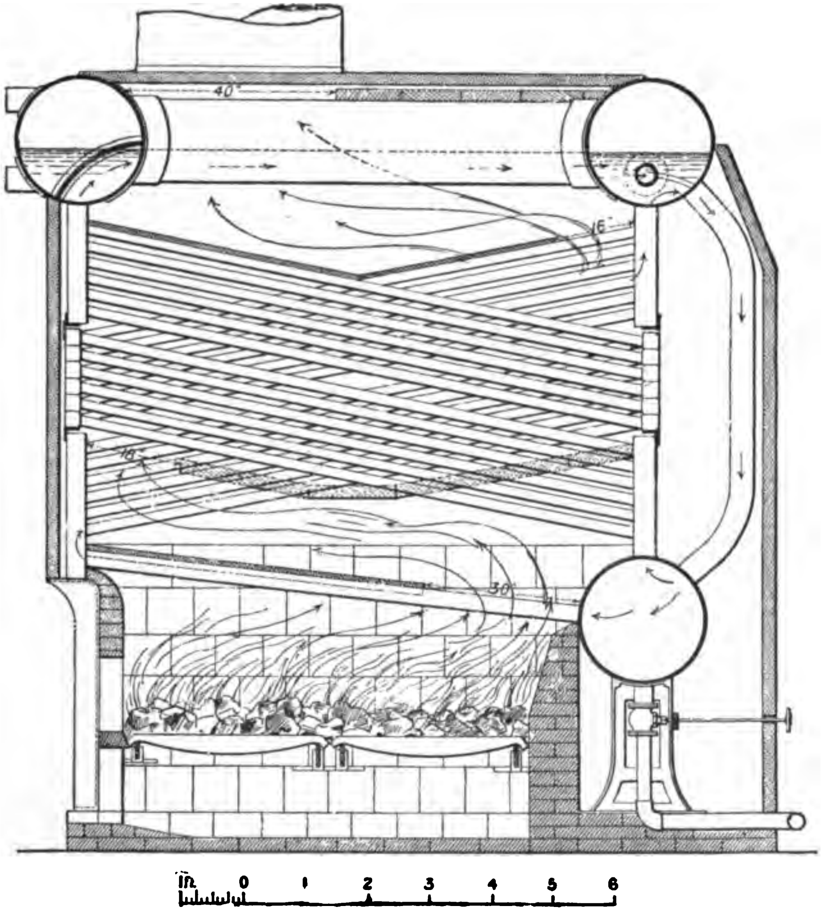


Fig. 155.—Hohenstein Boiler.

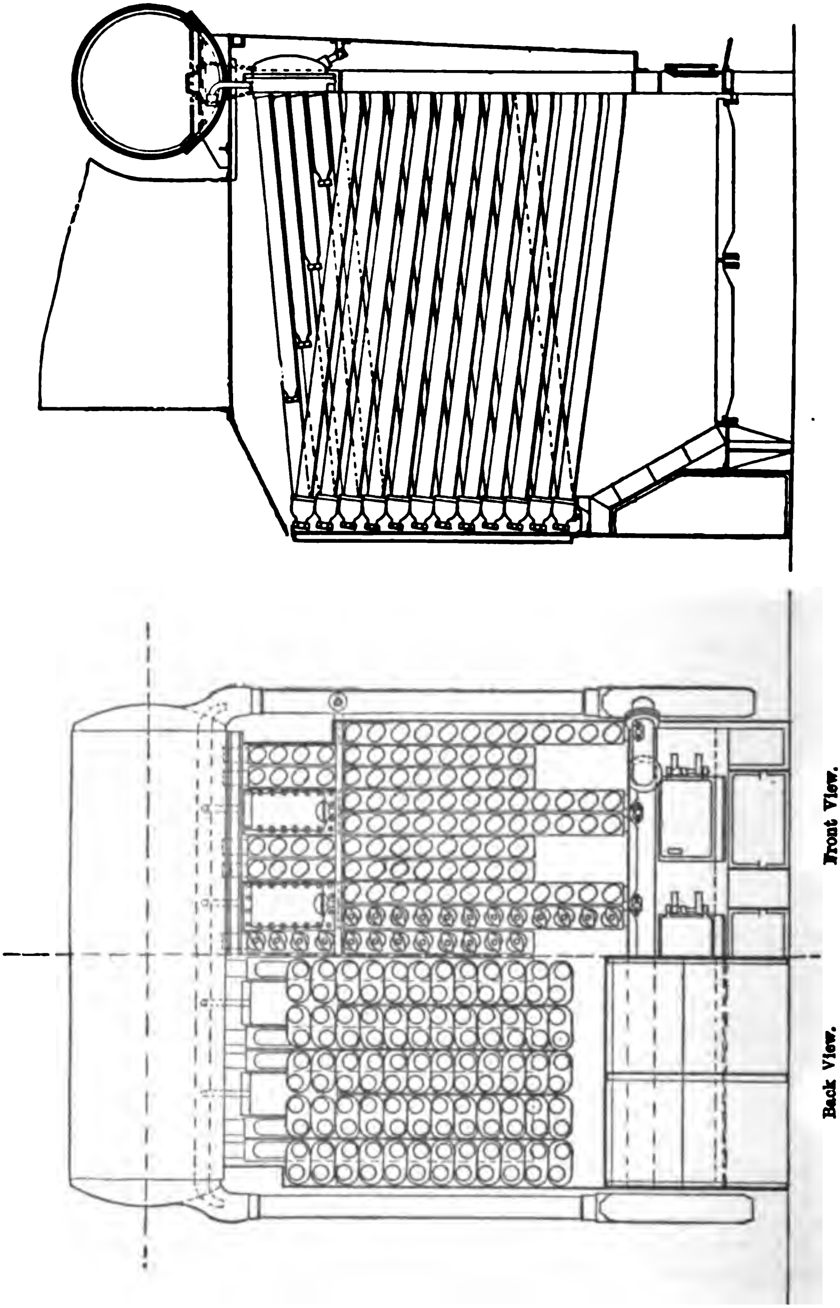


Fig. 156.—Thornycroft & Marshall's Boiler (Large Tube type).

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shown in fig. 156, like those in the Hohenstein. Here, too, the tubes are straight, and at a good angle; they can be readily examined and replaced or cleaned, and the circulation is certain and good. This boiler can also be readily drained and cleaned.

There are some other boilers worthy of notice in a special treatise, but not, however, of the same interest to the marine engineer as the foregoing.

The water in the water-tube boiler is, as a rule, very considerably less than that in the ordinary boiler, and to this fact is in no small measure due the exceedingly light weight of certain boilers of this type. To the designer this is of course a great advantage, and in case of accident, either from within or from without—from a shot, for example—the possible amount of harm is limited, especially if there be no obstruction with the draught; but to the firemen it is otherwise a source of trouble, as it permits of such great variations in steam pressure from temporary causes. The reducing valve, as fitted to the Belleville and other boilers, tends to prevent any ill effect from this being experienced in the engine-room, for, so long as the pressure in the boiler does not fall below that at which the apparatus is set to deliver, there will be little or no variation at the high-pressure valve-box.

Table xxxvii. shows among other things the proportion of water to the total weight in the various classes of boilers. The total weight includes, besides the boiler proper, the water, mountings and fittings; the furnace fittings, brickwork, &c.; the casings and uptakes to the funnel base; in fact, everything is included that is necessary to make a fair comparison. Taking naval boilers of the double-ended type, the average weight of water is about 29 per cent. of the total weight; while with single-ended naval boilers, it is about 26 per cent. The steam pressure for which these boilers were designed is, however, only 155 lbs. With double-ended boilers made in accordance with the Board of Trade Rules for pressures of 160 lbs., the water is about 33½ per cent. of the total weight, and about 32 per cent. with the single-ended boilers. With the gunboat type of boiler, the water is 33 per cent. when designed in accordance with the Admiralty Rules. With the Belleville boilers the water is only about 8 per cent. of the total weight; with the Babcock and Wilcox boilers about 14½ per cent.; with the Yarrow boiler about 15½ per cent.; with the Thornycroft boiler 15 per cent.; and with the Normand boiler about 24 per cent. The dry-bottom locomotive boiler, which is the rival of the smaller class of water-tube boiler, has water to the extent of 30 per cent.

It will be seen by reference to Table xxxvii. that the total weight per I.H.P. at forced draught is on the average with the ordinary naval double-ended boilers about 88 lbs., and with the single-ended ones 108 lbs. The lightest of all is 80·6 lbs. The Belleville and the Babcock and Wilcox are both under 80 lbs., and with an air pressure of 1 inch of water the weight is as low as 70 lbs. with the Belleville. Taking the naval cylindrical boilers with natural draught the average weight per I.H.P. is 121 lbs. with double-ended boilers, and 133 lbs. with single-ended. It is true that the latter figure is for the larger sizes now allowed by the Admiralty for battleships and cruisers. With the Belleville boilers the weight per I.H.P. is 82 lbs., and the Babcock and Wilcox 88·7 lbs.; and with the large allowance provided in the cruisers it is only 110 lbs., or 11 lbs. less than the old double-ended boilers for 155 lbs. pressure, and 23 lbs. less than the single-ended boilers for 155 lbs. pressure now supplied to the navy. It is manifest by these figures that there is a distinct saving in weight with the water-tube

TABLE XXX.—H.M.S. "HERMES." WATER CONSUMPTION TRIALS.

BELLEVILLE BOILERS. 300 LBS. WORKING PRESSURE. MADE IN MAY AND JUNE, 1899.

Description of Trial.	Per-centage of Full Power.	Steam Pressure.			Lbs. of Water per I.H.P. per Hour.			Lbs. of Coal per I.H.P. per Hour.	Remarks.
		Boilers.	Engines, Starboard.	Engines, Port.	Main Engines.	Auxiliary.	All Purposes.		
12 hours at 1,000 I.H.P.	10·0	241	169	176	21·9	5·2	27·1	2·5	Main links linked up as far as practicable. Cut-off in H.P. cylinder 13·5 per cent.
12 ,, 1,000 ,,	10·0	163	124	123	20·1	6·67	26·8	2·3	Main links in full gear—independent links run in. Cut-off in H.P. cylinders 47·8 per cent.
30 ,, 2,000 ,,	20·0	196	126	127	17·06	4·22	21·28	1·76	
30 ,, 7,500 ,,	75·0	263	223	222	15·3	2·1	17·4	1·57	
12 ,, 7,500 ,, (*Closed exhaust system in use.*)	75·0	264	228	219	...	...	16·84	1·5	Auxiliary exhaust at 26 lbs. pressure in use on evaporators and surplus to L.P. receivers.
8 hours at 9,000 I.H.P. (*Closed exhaust system in use.*)	90·0	270	249	244	...	...	17·13	1·54	Auxiliary exhaust at 29 lbs. pressure in use on L.P. receivers only.
8 hours at 10,000 I.H.P.	100·0	240	231	228	15·64	2·05	17·69	1·58	

Report of Boiler Committee.

TABLE XXXI.—H.M.S. "HIGHFLYER." RESULTS OF SPECIAL TRIALS. BELLEVILLE BOILERS.

Description of Trial.	Date.	Duration of Trial.	I.H.P.	Speed.	Revolutions.	Draught of Water.		Number of Boilers in Use.	Coal Consumption. per I.H.P. per Hour All Purposes.	Water.			Remarks, Defects, &c.
						Forward.	Aft.			Used from Reserve Tanks.	Distilled for Machinery.	Distilled for Ship.	
		Hours.		Knots.		Ft. Ins.	Ft. Ins.		Lbs.	Tons.	Tons.	Tons.	
A_1	1-3/1/1900	60	1190·5	9·8	86·35	*19 8	22 11	6	3·46	..	42	19·5	Two generator door joints, and 7 economiser door joints leaky. Two fusible plugs renewed. These defects caused 1 boiler to be out of use 4¾ hours.
A_2	3-6/1/1900	60	1205·5	9·9	85·9	to	to	7	3·38	..	46	12	Two economiser door joints leaky and 4 fusible plugs renewed. These leaky doors caused 1 boiler to be shut off and another substituted.
A_3	6-9/1/1900	60	1137·4	9·98	85·85	†18 9	21 6	8 decr. to 6	3·50	..	48	19	Joints of 2 doors leaky causing 2 boilers to be shut off.
A repeat.	14-17/2/1900	60	1101·9	9·82	85·89	20 0	22 2	5	3·16	..	16	14	No defects developed. This trial was a repetition of the three previous trials.
B_1	13-16/1/1900	60	2331·5	14·0	120·9	*20 6 to	22 5 to	12	2·90	..	44	10·5	Starboard evaporator feed pump rod and crosshead defective.
B_2	16-18/1/1900	60	3132·4	13·9	122·75	†18 8	21 7	12	2·87	6	68	21·0	Ten joints in main steam-pipe and many smaller joints leaky. Twelve fusible plugs renewed.
C_1	24-26/1/1900	60	5750·4	17·15	153·75	19 2	22 2	18	2·67	16	52	6·0	Crank shaft of P. distilling engine was broken on two occasions. S. crank pit pump, connecting and pump rods broken and barrel cracked. H.P. cylinders, 3 stays for supporting cover found broken after trial.
C_2	30/1/1900 to 1/2/1900	38	5707·8	17·05	152·6	19 11	22 6	18	2·58	14	44	..	Crank shafts of both distilling engines broken. S.I.P. crankhead bearing heated, causing curtailment of trial, and spare brasses were fitted.
C_2 repeat.	28/2/1900 to 3/3/1900	60	5907·0	17·0	152·05	20 0	22 6	15	2·37	..	45	..	End door of steam drum on No. 13 boiler leaky. Top of float box of No. 18 boiler defective. Several joints in main and auxiliary steam pipes leaky.
D_1	6-7/3/1900	30	7946·3	17·9	164·35	19 5	22 6	18	2·52	10	50	..	Impeller shaft of S. auxiliary circulating pump broken. A few joints in steam pipes leaky. Sixteen fusible plugs renewed.
D_2	10-11/3/1900	30	8760·9	18·97	170·48	18 0	21 11	18	2·58	43	50	..	Two joints in main steam pipe leaky. Impeller of S^d auxiliary circulating pump worked loose on shaft. Thirty fusible plugs renewed. On subsequent examination junk ring keep of P. A. L. P. piston was found broken.
E_1	17/3/1900	$12\frac{48}{60}$	9132·1	18·08	171·2	19 10	22 6	18	2·36	10	12	..	P.I.P. crankhead bearing heated, spare brasses fitted. One fusible plug renewed.
E_2	20/3/1900	$12\frac{13}{60}$	9268·2	18·71	171·55	18 8	22 4	18	2·47	10	12	..	P.H.P. and P.I.P. crank brasses worked warm. One fusible plug renewed.
F	24-27/3/1900	$65\frac{30}{60}$	1235·7	9·99	85·9	19 10	22 6	6	3·06	..	40	..	Bucket of P. air-pump loose on rod causing damage to webs of bucket and guards of valves. This caused the full-power trial to be discontinued after $1\frac{45}{60}$ hours.

NOTE.—On 9/2/00 after steaming 24 hours at 10 knots for the purpose of training the crew, the shafts of both auxiliary circulating pumps broke.

Report of Boiler Committee.

* Draught at commencement of trials.

† Draught at end of trials.

TABLE XXXII.—H.M.S. "MINERVA." RESULTS OF SPECIAL TRIALS. CYLINDRICAL BOILERS.

Description of Trial.	Date.	Duration of Trial.	I.H.P.	Speed.	Revolutions.	Draught of Water.				Number of Boilers in Use.	Coal Consumption per I.H.P. per Hour. All Purposes.	Water.			Remarks, Defects, &c.
						Forward.		Aft.				Used from Reserve Tank.	Distilled for Machinery.	Distilled for Ship.	
		Hours.		Knots.		Ft.	In.	Ft.	In.		Lbs.	Tons.	Tons.	Tons.	
A_1	1-3/1/1900	60	1083·8	9·84	66·5	19	10	23	7	3 & 4	2·88	...	5	15	
A_2	3-6/1/1900	60	1084·5	9·89	66·19	19	6	23	4	4	3·02	...	5	15	
A_3	6-9/1/1900	60	1084·95	9·98	65·98	19	1	23	1	4	2·99	...	5	15	
B_1	13-16/1/1900	60	3028·1	14·00	93·47	19	11	23	5	6	2·33	...	7·5	23	
B_2	16-18/1/1900	60	3169·0	13·93	94·2	18	11	23	2	7	2·24	...	10	22	
C_1	24-26/1/1900	60	6242·35	17·15	117·92	18	9	23	1	8	2·28	...	20	6	S.H.P. cylinder cover leaked badly.
C_2	30/1/1900 to 1/2/1900	38	6022·8	17·05	117·1	19	0	23	3	8	2·25	...	9	...	S.L.P. slide rod gland leaked badly. Trial curtailed on account of defects in "Highflyer."
C_2 repeat.	28/2/1900 to 3/3/1900	60	6021·7	17·00	117·3	19	2	23	1	8	2·22	...	15	...	No defects. This trial is a repeat of two former trials.
D_1	6-7/3/1900	30	7550·2	17·53	123·9	19	3	23	2	8	2·23	20	24	...	Forward main feed suction pipe burst. S.H.P. piston-rod gland leaked badly.
D_2	10-11/3/1900	30	7379·2	18·03	123·6	18	2	22	11	8	2·16	...	12	...	Starboard main feed suction pipe burst.
E_1	17/3/1900	13[illegible]	6778·16	16·98	119·8	19	6	23	3	8	2·16	...	2	...	
E_2	20/3/1900	13[illegible]	6510·3	16·83	117·3	19	0	23	0	8	2·11	...	2	...	
F	24-27/3/1900	65[illegible]	1132·0	10·00	66·1	19	10	23	3	3 to 8	3·09	...	5	...	No defects. Trial at full power discontinued at end of 1[illegible] hours on account of defects in "Highflyer."

Report of Boiler Committee.

TABLE XXXIII.—Particulars of Machinery of "Minerva" and "Highflyer."

Name of Vessel, - - -	"Minerva."	"Highflyer."
Engines—		
Type, - - - - - -	Triple expansion (3 cylinders).	Triple expansion (4 cylinders).
I.H.P. maximum, - - - - -	9600	10,000
Revolutions, - - - - - -	140	180
Stroke, - - - - - - -	3′ 3″	2′ 6″
Boilers—		
Type and number, - - -	Tank—8 in No.	Belleville, 18 in No.
Steam pressure, - - -	155 lbs. at boilers, 150 ,, engines.	300 lbs. at boilers, 250 ,, engines.
Weight—		
Main and auxiliary machinery, -	380·1 tons.	373·8 tons.
Boilers, - - - - - - -	557·8 ,,	461·5 ,,
Total (ready for steaming), - -	937·9 ,,	835·3 ,,

Report of Boiler Committee.

TABLE XXXIV.—Trials of Ships of "A" Fleet (Manœuvres, 1899), Showing Coal Consumption on Service with various Boilers. ⅗ N.D. Power.

Name of Vessel, - -	"Retribution."	"Arethusa."	"Furious."	"Pactolus."
Boilers, - - - - -	Cylindrical D.E.	Cylindrical	Belleville.	Blechynden.
Pressure steam load on safety valves, - - lbs.	155	75	250	300
Engines, - - - - -	Triple.	Compound.	Triple.	Triple.
Total distance run, - miles	44	43	49	48
Duration, - - - hours	3	3	3	3
I.H.P., - - - - - -	4399	2715	6393	2923
Proportion of authorised natural draught power, per cent.	62·8	54·3	63·9	58·5
Speed, - - - - - knots	14·7	14·3	16·33	16·0
Revolutions per minute, No.	107·0	78·0	122·52	166·3
Consumption of coal per I.H.P. per hour, all purposes, - - - lbs.	1·98	2·8	2·45	2·91
Description of coal, - -	Naval Merthyr.	Hood's Merthyr.	Cory's Merthyr.	Cambrian Navigation.
Quality of coal, - - - -	8 to 9	Good.	7 to 8	7
Total coal used, - - tons	11·7	10·0	21·0	11·4

Report of Boiler Committee.

TABLE XXXV.—H.M.S. "DIANA." WATER CONSUMPTION TRIALS, SHOWING COST OF AUXILIARIES.
CYLINDRICAL S.E. BOILERS. 155 LBS. WORKING PRESSURE.
MADE IN MAY AND JUNE, 1899.

Description of Trial.	Percentage of Full Power.	Steam Pressure.			Lbs. of Water per I.H.P. per Hour.			Lbs. of Coal per I.H.P. per Hour.	Remarks.
		Boilers.	Engines, Starboard.	Engines, Port.	Main Engines.	Auxiliary.	All Purposes.		
30 hours at 800 I.H.P.	8·3	101	96	97	20·59	5·68	26·27	2·82	
12 „ 1600 „	16·6	124	120	121	18·94	4·21	23·15	2·52	Cut-off in H.P. cylinder 38 per cent.
30 „ 1600 „	16·6	117	113	113	18·26	3·29	21·55	2·52	Cut-off in H.P. cylinder 44·5 per cent.
30 „ 4800 „	50·0	133	128	128	17·09	2·14	19·23	2·02	
30 „ 6400 „	66·6	138	129	129	17·9	2·56	20·46	2·38	
8 „ 8000 „	83·3	150	144	144	17·75	1·94	19·69	2·39	

Report of Boiler Committee.

TABLE XXXVI.—Comparison of Water Tube Boilers with Others in Various Ships (Trial Trips).

Class of Ship, - -	Battleships.		First-Class Cruisers.		Second-Class Cruisers.			Third-Class Cruisers.		Torpedo Gunboats.			
Name, - - -	Mean of Six.	Mean of Six.	"Edgar" Class.	"Diadem" Class.	"Diana" Class.	"Apollo" Class.	"Arrogant" Class.	"Pearl" Class.	"Pelorus" Class.	"Alarm" Class.	"Speedy."	"Sheldrake."	"Seagull."
Indicated horse-power, -	12,414	13,500	12,851	16,961	9,846	9,274	10,240	7,469	7,152	3,665	4,703	4,050	3,546
Boiler pressure, - -	149	289	146·2	286·5	151	155	265	155	253	155	...	151	145
Air pressure in stokehole,	0·85	0·19	0·7	0·305	1·06	0·9	0·06	1·4	2·9	2·20	1·70	0·5	0·16
Type of boiler, -	Cylindrical.	Belleville.	Cylindrical.	Belleville.	Cylind. S.E.	Cylind. D.E.	Belleville, old.	Cylind. D.E.	Express.	Locomotive.	Thornycroft.	Babcock.	Niclausse.
Grate area, - - sq. ft.	817	1,050	812	1,460	624	575	869	410	353	171	204	252	276
Total heating surface, ,,	25,233	33,700	24,908	40,990	18,579	15,641	25,606	11,025	20,508	6,204	17,700	9,103	7,932
Total weight of boilers, &c., tons	724	623	646	767	548	434	463	314	172	124·4	132	124·8	1,347
Ratio heating surface to grate, - - -	30·8	32·1	30·7	28·1	29·8	27·2	29·4	26·9	58·0	36·3	86·8	37·4	28·8
Coal per I.H.P. per hr., lbs.	2·29	1·72	1·70 (*a*)	1·85	2·18 (*b*)	...	2·00 (*c*)	...	2·36 (*d*)	...	...	1·57	2·11
,, sq. ft. grate per hour, - - ,,	34·8	...	...	...	...	...	...	...	...	...	...	25·6	27·1
Weight of boilers, &c., per I.H.P., - - ,,	131	103	111	101	124	104	99	98·6	52·9	76·0	62·9	69·9	85·1
Weight of boilers, &c., per sq. ft. of grate, ,,	1,971	1,329	1,687	1,172	1,959	1,674	1,170	1,715	1,092	1,630	1,450	1,109	1,093
I.H.P. per sq. ft. of grate,	15·2	12·9	15·2	11·6	15·79	16·1	11·82	18·2	20·9	21·4	23·0	16·0	12·8
,, ton of boiler, -	17·1	21·7	20·1	22·1	17·97	20·7	22·5	23·8	43·1	29·5	39·6	32·5	26·3
Total heating surface per I.H.P., - -	2·0	2·5	1·9	2·41	1·88	1·65	2·49	1·47	2·87	1·70	3·7	2·3	2·24
Total weight of machinery, - tons	1,341	1,290	1,161	1,540	915	740	825	539	383	211	212	205	2·14
I.H.P. per ton of machinery,	9·3	10·5	11·1	11·0	10·76	12·5	12·34	13·8	18·92	17·3	22·1	19·3	16·9
Piston speed, - - ft.	875	918	923	923	937	910	906	880	981	875	875	875	...
Coal consumption per I.H.P. at sea trial,	2·0	2·36	2·0	2·33	2·11	...	2·36	...	...	...	...	2·14	2·15
I.H.P. on sea trial, 1 ship,	8,029	8,163	8,824	13,517	6,694	...	6,273	...	...	...	...	2,775	2,773

(*a*) At 10,517 I.H.P., (*b*) at 8307 I.H.P., (*c*) at 7275 H.P., (*d*) at 5388 H.P.

TABLE XXXVII.—Weights of Boilers of Various Kinds.

Name.	Description.	Pressure.	Total Weight.	Weight of Water.	Indicated Horse Power.		Weight per I.H.P. in Lbs.	
					Natural Draught.	Forced Draught.	Natural Draught.	Forced Draught.
		Lbs.	Tons.	Tons.				
H.M.S. "Immortalite,"	Double-ended Cylindrical,	135	188·5	59·2	3,490	4,680	118	88·0
,, "Andromache,"	,, ,, ,,	155	195·0	53·4	3,800	5,010	115	80·6
,, "Endymion,"	,, ,, ,,	155	278·0	87·3	5,425	6,100	115	102·0
,, "Pearl,"	,, ,, ,,	155	150·0	39·6	2,770	4,080	121	82·7
,, "Blanche,"	,, ,, ,,	155	117·5	33·3	1,940	3,000	136	87·8
,, "Majestic,"	Single- ,, ,,	155	325·0	86·0	5,000	6,000	145	121·3
,, "Charybdis,"	,, ,, ,,	155	205·0	54·3	3,805	4,850	121	94·7
,, "Magpie,"	Gunboat Type ,,	145	53·5	18·7	1,030	1,340	116	89·4
,, "Terrible,"	Belleville Water-tube,	260	888·0	66·0	18,000	25,000	110	79·6
Foreign Cruiser.	,, ,, ,,	260	548·0	48·0	15,000	17,500	81·9	70·1
,, ,,	Babcock & Wilcox Water-tube,	200	594·0	87·0	15,000	17,500	88·7	76·0
,, ,,	Double-ended Cylindrical,	175	697·0	206·5	15,000	17,500	104	89·2
H.M.S. "Hornet,"	Yarrow Water-tube,	180	5·4	0·6	...	487	...	24·8
,, "Salmon,"	,, ,, ,,	180	10·25	1·6	...	916	...	25·0
Torpedo Boat,	,, ,, ,, "Cintra,"	180	9·6	...	...	875	...	24·5
H.M.S. "Decoy,"	Thornycroft Water-tube,	210	42·55	6·3	...	4,570	...	20·9
,, "Ferret,"	Normand ,, ,,	175	47·6	11·46	...	4,510	...	23·6
,, "Conflict,"	White ,, ,,	185	61·45	...	...	4,931	...	...
,, "Havock,"	Locomotive Dry Bottom,	180	53·0	...	...	3,600	...	33·0
S.S. "Cazengo,"	Double-ended Cylindrical, Board of Trade,	160	201·7	76·0	3,020	...	149	...
,, "Gothan,"	,, ,, ,, ,,	210	105·0	31·2	1,200	...	196	...
,, "Vienna,"	Single- , ,, ,,	166	212·6	63·6	2,616	3,366	179	141
,, "Victoria,"	,, ,, ,, ,,	150	110·5	32·0	1,540	...	161	...
,, "Polo,"	,, ,, ,, ,,	200	50·7	...	500	...	227	...
,, "Egyptian M,"	Belleville Water-tube,	250	81·0	6·0	1,700	...	107	...
,, "Hero,"	Babcock & Wilcox,	200	41·0	...	800	...	115	...

boilers in large ships. It is, however, in the smaller class of vessels that the great advantage in using these boilers is experienced. Here types of boilers can be employed that would hardly be admissible in the larger ships, and the full advantage of the system can be utilised. By again referring to the table, it will be seen that with the locomotive boiler the weight per I.H.P. is 33 lbs. with the single-ended variety, and 30·4 lbs. with the double-ended. With the Yarrow boiler the weight is only 25 lbs., and considerably under the locomotive, as are all the other forms now fitted into torpedo boat destroyers. The great saving in each case is, of course, due in great measure to the water. Referring to Table xxxvi. it will be seen that, when taking all the weights which come into and are included in the boiler-room weights according to Admiralty Rule, for each ton with single-ended boiler, &c., 17·53 I.H.P. was developed by the engines; with double-ended boilers, 20·1 to 23·8; with Belleville boilers, 21·7 to 22·5; with Thornycroft, &c., in 3rd class cruisers, 43·1; with locomotive in torpedo gunboats, 29·5; with Babcock and Wilcox, in same ships, 32·5; with Niclausse, 26·3. With the express boilers in torpedo boat destroyers of recent build as much as 95·9 I.H.P. has been developed with Thornycroft's, 84·6 with Normand's, 79·1 with Yarrow's, and 70·6 with Reed's, as against 60 with locomotives in the "Havock."

When the cylindrical boiler is designed in accordance with the Board of Trade Rules, the difference is more marked, for here the double-ended boiler is such that the total weight per I.H.P. is 149 lbs. when the working pressure is 160 lbs., and as much as 196 lbs. when it is 210 lbs. pressure. Single-ended boilers are, of course, heavier still. The Belleville boiler is only 107 lbs. per I.H.P. with a working pressure of 250 lbs., and the Babcock and Wilcox 115 lbs. for a pressure of 200 lbs.

The space occupied by these boilers does not differ materially from that required for cylindrical or locomotive boilers, and the stoke-holes, as a rule, must be as long; but with some water-tube boilers a comparatively short stoke-hole is sufficient. The water-tube boiler is capable of a subdivision that does not obtain with the cylindrical, and in this respect it is in the hands of the designer much as the old box boiler was.

Feed Arrangements.—It is, of course, essential to the good working of these boilers, as of all boilers containing such comparatively small quantities of water, that the supply of feed-water shall be regular and delivered in such a way as to improve rather than to retard the circulation. To satisfy the first condition, it is necessary to have good feed-pumps and a good supply of water, from which they may draw whenever required. The ordinary engine pump drawing from the hot-well is always more or less intermittent in its action, and while no great inconvenience is felt when it is supplying the ordinary tank boiler or the larger kinds of water-tube boiler, it would not suit the more delicate ones. It may, therefore, be taken, as a rule, that this class of boiler should have an automatic apparatus for controlling the feed. The Belleville boiler is always fitted with an arrangement, which consists essentially of a chamber in which there is no disturbance by circulation or other cause, but so connected to the boiler that the water level in it is the normal level of the water in the boiler. It contains a "float" arrangement which operates in an ingenious, if somewhat complicated, way, a controlling valve on the feed-pipe, so that, as the water falls in the boiler, the feed-water is allowed to pass, and as the water rises above the working level it is gradually shut off. Sir J. I. Thornycroft and others have adopted somewhat similar methods for regulating the supply. Other engineers, including Mr. Yarrow, have, however, preferred to fit to each boiler an independent donkey pump, and to set it working with steam from its own boiler at such a speed as to just keep up the supply. This method, while convenient, cannot be said to be automatic; it might, however, be so if to the feed-valve, or even to the steam-valve, of the donkey is fitted a means of controlling by a similar apparatus to the above. As the variation of level in most water-tube boilers is somewhat limited, the "float" apparatus cannot have a great range.

Mr. Yarrow has adopted an ingenious arrangement for controlling the feed donkeys which is at the same time very simple and effective. There is an internal steam pipe in the steam receiver specially fitted for the feed donkey; its open end is surrounded by a small trough, and is at the highest working level of the water. (The trough is shaped like an inverted cone, and so keeps the water fairly smooth within it.) If the water rises above this point, water enters the steam pipe, gags the pump, and is carried through the exhaust pipe to the condenser or hot-well tank.

It is a disputed point whether to admit the feed-water into the upper or lower chamber; but if the principle of aiding and not obstructing circulation is observed, there need be little or no controversy on this point. Each case must be dealt with on its own merits; but, as a general rule, unless the feed-water is well heated, it is better admitted to the bottom chamber or at the downcast part of the upper chamber in a direction of the downward flowing currents; by preference the latter is more likely to aid circulation. In this case, the feed-pipe may with advantage be made small, so that the water is injected at a high velocity. When the water is well heated before admission to the boiler proper it is not of much importance where it is taken, but an internal pipe with small outlets to insure distribution should be fitted.

It is usual and advisable also to fit water-tube boilers with a settling chamber into which any solid matter admitted with the feed-water can be deposited and blown out. The principle on which these act is that solid matter of higher specific gravity than the water will deposit in an enlargement of a channel where there is a reduction in the velocity of flow; a change in direction of flow also helps to deposit. A certain amount of lime is used with advantage in these boilers, and a means for admitting it is necessary. As a rule, it may be introduced with the feed-water by placing it in the tanks.

In arranging casings and uptakes great care should be take to insure a good distribution of the hot gases over the whole of the tubes. The tendency to shorten circuit is, of course, strong, and to prevent it considerable judgment is necessary in the placing of baffle plates and other devices. It is a matter of experiment, however, to so place them as to get the best results.

The fire-places in water-tube boilers are generally of such a nature as to require a brickwork erection; in fact, in the case of the Belleville boiler the brickwork is very extensive, but in the case of most of the others, and especially of the lighter kinds, the brickwork is limited to the immediate neighbourhood of the fire, the rest of the boiler up to the base of the funnel being enclosed with light ironwork, either lined with asbestos or in some other way protected from the direct action of the heat of the gases. As far as possible it is better to use water walls than brickwork, both of which are only absolutely necessary to resist contact of the flame or direct radiation from the glowing fuel. Casing plates outside the tubes, not subject to the direct action of the fire, may be of very thin sheet steel, especially if stiffened with light angles or by corrugations; it is generally made portable, so that it can be easily removed for the cleaning of the outside of the tubes or the repairing of the boiler when necessary. Sheet asbestos ¼ inch thick forms a very good protection to these thin casings; it requires to be carefully secured and protected from damage, and for this purpose it is often completely covered by thin sheet steel, but this covering should be avoided if possible. Two layers of asbestos millboard with a space between them practically sealed, so as to prevent air circulation, makes a still better shield, and with care in fixing it will withstand a considerable amount of wear and tear. The Admiralty do not now require the fitting of dampers to the uptakes of water-tube boilers as fitted in the torpedo boat destroyers, and, indeed, the necessity for them does not exist, while their presence may be a source of danger. The evaporation with these boilers is quickly checked by leaving the fire door open, or, when so arranged, by shutting off altogether access of air to the fire. Without the damper the bursting of a tube or other serious leakage is not dangerous, as the steam escapes immediately up the funnel. The Admiralty also require to be fitted an apparatus for drenching the fires with water, which can be operated both from the stokehole and on deck. The tubes do not get covered with soot and dust so badly as might be expected, and when Welsh coal is used in the boilers the deposit is easily removed.

At present the life of these water-tube boilers is too short to be satisfactory, and the trouble experienced with some, especially the Belleville, due in no small measure to the want of care and judgment on the part of those attending them, has caused them to be condemned. Others less in first cost, and, owing to not having such delicate fittings and casings, &c., not requiring so much care and attention, are likely to be more generally used for both naval and mercantile ships. It is, however, imperative for success in either service that the tubes last at least as long as those of a cylindrical boiler; that they be straight, or nearly so, and inclined at a good angle with the horizon—say, at least, 8 to 10 degrees; that the circulation be certain, uniform, and rapid; that the surfaces exposed to heat be easily cleaned and kept clean; that the fittings be as few in number as possible and absolutely tight; that the casings be so designed as to remain airtight and to keep good as long as the boiler itself, and that the grates be of such a size and so placed as to get good combustion and distribution of the heat.

TABLE XXXVIII.—Trials of Various Boilers (Marine).

Type of Boiler.	Ratio Total Heating Surface to Grate.	Duration of Trial.	Kind of Fuel.	Fuel per Hour.		Ash.	Temperature. Feed Water.	Steam Pressure.	Air.		Water Evaporated.		Equivalent from and at 212°.			A × B.	Remarks.
				Total.	Per Sq. Ft. of Grate.				Pressure.	Temperature.	Per Hour.	Per Lb. of Fuel.	Per Hour.	Per Lb. of Fuel.	Per Sq. Ft. of Heating Surface		
		Hrs.		Lbs.	Lbs.	%.	° F.	Lbs.	Ins.					A.	B.		
Cylindrical, Single End,	22·3	3·75	Welsh.	418	19·8	2·81	61	70	...	...	3,190	7·64	3,800	9·09	7·60	69·1	Coal-testing trial.
Locomotive, -	70·0	...	,,	440	35·5	5·70	...	130	...	...	...	...	5,370	12·20	6·20	75·6	By Donkin & Kennedy.
Belleville (original),	28·5	8·0	,,	2623	19·4	...	50	194	...	...	...	9·07	29,115	11·10	7·56	83·9	Contract quantity trial.
,, ,,	28·8	4·0	,,	1309	27·3	...	42	275	0·75	...	...	7·60	12,305	9·40	8·63	81·1	,, ,,
Niclausse, -	34·5	7·27	Pwll Dffrn.	254	13·5	6·74	60	158	0·20	67·4	...	8·67	2,659	10·47	4·09	...	By Profs. Kennedy and Unwin.
,,	34·5	2·95	,,	482	25·6	6·74	55	144	0·56	62·3	...	8·53	4,984	10·34	7·67	79·3	,, ,,
Babcock & Wilcox,	37·4	3·0	,,	1260	20·0	...	115	200	0·10	...	11,483	9·11	13,784	10·94	5·85	...	Admiralty trial.
,, ,,	37·4	3·0	Nixon's Nav.	2564	40·7	...	70	185	0·25	...	19,577	7·63	23,460	9·15	10·00	91·5	,, ,,
Stirling, -	48·5	5·0	Welsh.	524	16·0	6·98	102	198	...	...	5,433	10·30	6,314	12·05	3·95	...	
,, -	48·5	7·5	,,	700	21·0	7·00	100	195	...	...	6,600	9·47	7,756	11·08	4·85	...	
,, -	48·5	4·0	,,	994	30·1	8·00	100	200	...	...	8,857	8·92	10,377	10·44	6·49	67·7	
Haythorn, -	63·0	...	,,	102·6	15·1	...	47·2	196	Nat.	60·0	909	8·86	1,114	10·86	2·60	...	By Prof. Watkinson.
,, -	63·0	...	,,	271·3	39·8	...	52·0	240	2·0	80·0	2,565	9·45	3,147	11·60	7·35	85·3	,, ,,
,, -	63·0	...	,,	504	74·1	...	55·6	251	2·6	61·0	3,648	7·24	4,486	8·90	10·48	93·3	,, ,,
Express :—																	
Reed's, -	67·6	...	,,	1684	19·1	...	...	...	Nat.	...	...	...	7,969	11·60	2·30	...	
,, -	33·8	...	,,	3344	46·6	...	...	...	1·5	...	...	...	31,551	9·42	9·20	86·7	
,, -	33·8	...	,,	...	79·4	...	...	...	3·5	...	...	...	47,217	8·28	13·80	114·2	
Thornycroft,	49·0	3·0	,,	3024	67·2	...	56·0	225	3·0	50·0	21,295	7·04	25,976	8·59	12·20	107·8	Contract quantity trial.
Mumford's, -	45·0	4·0	,,	...	25·0	...	56·0	175	0·15	...	...	9·00	...	10·90	...	...	Admiralty trial.
Seaton's, -	25·8	6·0	,,	178	17·45	3·27	56·0	200	0·32	...	1,356	7·61	1,648	9·26	6·27	58·1	Coal test.
,, -	25·8	3·0	,,	461	45·5	1·50	56·0	200	2·00	...	3,237	7·02	3,900	8·46	14·83	125·5	,,
Yarrow, -	55·7	...	,,	627	19·0	...	...	180	...	...	...	...	7,524	12·00	3·00	...	Prof. Lambert and Mr. Haddon.
,, -	55·7	...	,,	1716	52·0	...	...	180	...	...	...	...	17,675	10·30	9·50	97·9	,, ,,

CHAPTER XX.

BOILERS—CONSTRUCTION AND DETAILS.

BOILERS for the mercantile marine are nearly always constructed in accordance with the rules laid down for the scantlings by the Board of Trade, Lloyd's or other Registry. If a ship requires a passenger certificate, the boiler must be built in accordance with the Board of Trade rules and regulations; and if to class at Lloyd's, those of Lloyd's Registry must be followed. If both a Board of Trade certificate and a Lloyd's machinery certificate are requisite, the boiler must be so designed as to accord with the rules of both. It is much to be regretted that these two public institutions do not agree to one set of rules which shall be acceptable to both, and in accordance with the best engineering knowledge and experience of the day. Both sets of rules are based on scientific principles and practical experiments on a large scale, and differ only slightly from one another in consequence. They chiefly differ on the question of the factor of safety, and the differences on this point arise from the allowance included covertly for wear and tear; the survey in the one case being yearly, and in the other often at longer intervals.

Ships built for mercantile purposes, but neither classed at a Registry nor requiring a passenger certificate, have boilers constructed *in accordance with the rules* of one or the other institution; and he would be a bold man who should choose to depart much from either, and be responsible for it.

The Admiralty rules are not so stringent nor so extended as those of the Board of Trade, as the circumstances of naval construction require a little more elasticity.

Boiler Shell, Cylindrical.—This is the simplest and strongest form to withstand internal pressure; because, since a circle is the figure of least perimeter for a given area, there is no tendency to change of form, the metal is strained in one way only—viz., tangentially and in tension.

The total pressure tending to rupture a cylinder is the part of all the pressures acting at the various points in direction normal to the surface resolved in one direction. This is equivalent to the pressure on the plane through the axis of the cylinder.

Rupture is resisted by the two sections of metal at the sides.

Let D be the diameter of the thin cylinder, and L its length in inches; p the effective pressure per square inch, and t the thickness of metal in fractions of an inch.

Then the pressure tending to burst it $= p \times D \times L$.

The stress per square inch on the metal resisting this is

$$= \frac{p \times D \times L}{L \times 2t} = \frac{p \times D}{2\,t}.$$

Let T be the ultimate strength of the material in pounds per square inch, and F be the factor of safety deemed advisable, and let $T \div F = f$.

TABLE XXXIX.—Tests of Boiler Material (Steel).

Part of Boiler.		Admiralty.		Board of Trade.		Lloyd's Register.		British Corporation.		Bureau Veritas.	
		Ultimate Strength.	Exten-sion.	Ultimate Strength.	Exten-sion.	Ultimate Strength.	Exten-sion.	Ultimate Strength.	Exten-sion.	Ultimate Strength.	Exten-sion.
			Per cent.		Per cent.		Per cent.		Per cent.		Per cent.
Shell plates,	Tons per sq. in.	27 to 30	20	27 to 35	18	28 to 32	20	28 to 32	20	to 30	20
Plates exposed to flame,	,,	24 to 27	25	26 to 30	20	26 to 30	23	26 to 30	23	to 28	22
Rivet bars,	,,	24 to 27	25	26 to 30	25	26 to 30	25	26 to 30	25	24 to 27	25
Stay bars,	,,	27 to 30	20	27 to 32	20	28 to 32	20	28 to 32	20	to 30	20
Steam-pipe plates,	,,	24 to 27	33	...	...	...	...	...	...	...	...
Corrugated furnaces,	,,	23 to 25	27	...	...	26 to 30	23	26 to 30	23	to 25	25
Tubes, ...		26	27	26 to 30	20	...	...	...	...	22 to 25	22

Then the safe working pressure for a boiler shell, or other cylindrical part subject to internal pressure $= \frac{2t \times f}{D}$.

This holds good only when there is no joint or other cause of reduction of effective area of plate section.

Since a boiler shell is made of one or more plates connected by riveted joints, the effective area of plate is that part remaining between the rivet holes (neglecting the effect of the friction between the plates).

If the plates are connected by means of a single row of rivets, the average value of the part remaining between the rivet holes is generally 56 per cent. of the whole plate; so that in this case

$$\text{Safe working pressure} = \frac{2t \times f}{D} \times \frac{56}{100} = \frac{t \times f}{D} \times 1{\cdot}12.$$

When the plates are connected by two rows of rivets, so that the joint is said to be a double-riveted one, there is on the average 70 per cent. of the plate remaining between the holes; in this case

$$\text{Safe working pressure} = \frac{2t \times f}{D} \times \frac{70}{100} = \frac{t \times f}{D} \times 1{\cdot}40.$$

It will be seen that the double-riveted joint is more than 25 per cent. stronger than the single-riveted. This difference is due to the wider spacing of the rivets which can be allowed, in consequence of the increase of rivet area obtained with the two rows.

The value of F depends on the kind of material, and on its quality.

The Board of Trade requires that the steel used in the shells of boilers shall have a minimum strength of 27 tons and a maximum of 32 tons per square inch, with an elongation of not less than 18 per cent. in 10 inches, and allows 28 tons to be used for calculations. If the plates stand a test of over 28 tons, that figure may be taken for calculation purposes. Steel up to 35 tons has been used for boiler shell, and is likely to be so generally. The Board, however, requires a minimum factor of safety of 4·5, while the Admiralty are content with 4·0, which is quite sufficient, considering that the stress on the material is gently applied and quite a steady one during work.

Table xxxix. gives the tests of the materials required by the Admiralty, Board of Trade, Lloyd's Register, &c.

Riveting.—The longitudinal seams or joints of a boiler shell are made in one of the following ways:—

(1) *Lap joint and single-riveted.*—The plates in this case are lapped one over another, and united by a single row of rivets. This method is only adopted when the plate is thicker than required for mere strength, so that 56 per cent. of it is sufficient to safely withstand the pressure. This generally arises in the case of small boilers and steam receivers, where the diameter is comparatively small, or with low pressures, when the thickness of plate which is sufficient to withstand the pressure would not permit of caulking, nor give the allowance for deterioration.

Rivet metal is always softer than boiler plate, and its resistance to shearing is less than that of the latter to tension, for this reason the area of the rivets should be greater than that of the plate remaining between the rivet holes; but, on the other hand, whereas the plate is subject to reduction by wear, the rivet section is not so affected, and in consequence it might suffice to allow the same area of rivet. The Board of Trade requires the rivet area of section to be $\frac{28}{23}$, or 1·22 times the area of section of plate between the holes.

Taking 56 per cent. as the proportion of plate between the holes, p the pitch, and d the diameter of the rivets,

$$\frac{\text{Pitch of rivets}}{\text{diameter}} = \frac{100}{100 - 56},$$

or

$$\text{Pitch of rivets} = 2{\cdot}273 \times \text{diameter}.$$

Since the area of rivet = the portion of the plate between holes,

$$\frac{\pi}{4} d^2 = \frac{28}{23} (p - d) t = \frac{56}{100} p \times t \times \frac{28}{23}.$$

Substituting the value of p as found above,

$$\text{Diameter of rivet} = 1{\cdot}97 \times \text{thickness of plate}.$$

Example.—To find the pitch and diameter of rivets for a single-riveted lap joint, with plates ½ inch thick, strength of joint being 56 per cent.

Diameter of rivet = 1·97 × ½, or 0·98 inch.
Pitch of rivets = 2·273 × 0·98, or 2·23 inches.

The lap of the plates is three times the diameter of the rivet; if it is made more than this, the plate will spring in caulking; if made less, there is danger of bulging or cracking the edges, and also there is no margin for recaulking if required.

The following table gives the pitch, &c., as found in general practice :—

TABLE XL.—LAP JOINTS, SINGLE-RIVETED, TIGHT WORK.

Thickness of Plate.	Diameter of Rivet.	Pitch of Rivet.	Breadth of Lap.	Percentage of		Board of Trade P. C. Joint.
				Plate.	Rivet.	
¼	½	1 3/16	1½	58·0	66·1	54·2
5/16	⅝	1½	1⅞	58·3	65·4	53·6
⅜	¾	1¾	2¼	57·1	67·3	55·2
7/16	⅞	2⅛	2⅝	58·8	65·2	53·4
½	1	2¼	2⅞	55·6	70·0	55·6
9/16	1	2 5/16	3	56·7	60·0	50·0

(2) *Lap joint and double-riveted.*—Here there are two rows of rivets. Sometimes, as in shipbuilding, the rivets of one row are exactly in line with those of the other, and thus called "chain" riveting; but more frequently those of one row are in line with the middle of the spaces of the other, and it is then called "zigzag" riveting. The latter plan requires less lap, and makes tighter work than the former, but does not leave the plate so strong, especially when the holes are punched. Here again the rivet area may be

made the same as the area of the plates between the holes, but the Board or Trade requires the area of rivet section to be $\frac{28}{23}$ or 1·22 that of the plate between the rivet holes. Taking the proportion of the plate between the holes at 70 per cent.

$$\frac{\text{Pitch of rivets}}{\text{diameter}} = \frac{1000}{100 - 70},$$

or, Pitch of rivets = 3·333 × diameter.

Also,
$$2 \times \frac{\pi d^2}{4} = \frac{70}{100} p \times t \times \frac{28}{23} = 0{\cdot}8\, p \times t.$$

whence, substituting the above value of p,

Diameter of rivet = 1·7 × thickness of plate.

Example.—To find the pitch and diameter of the rivets for a lap joint, double-riveted; the plates being $\frac{3}{4}$-inch thick, and the strength of joint 72 per cent.

Diameter of rivet = 1·7 × $\frac{3}{4}$, or 1·275 inches.
Pitch of rivet = 3·333 × 1·275, or 4·35 inches.

The following table gives the pitch, &c., as found in general practice:—

TABLE XLI.—LAP JOINTS, DOUBLE-RIVETED (ZIGZAG).

Thickness of Plate.	Diameter of Rivets	Pitch of Rivets.	Breadth of Lap.	Percentage of		
				Plate.	Rivets.	B. T. Joint.
$\frac{7}{16}$	$\frac{3}{4}$	$2\frac{1}{2}$	$3\frac{3}{4}$	70·0	80·8	66·2
$\frac{7}{16}$	$\frac{13}{16}$	$2\frac{3}{4}$	4	70·4	86·2	70·4
$\frac{1}{2}$	$\frac{7}{8}$	3	$4\frac{1}{4}$	70·8	80·1	65·7
$\frac{1}{2}$	$\frac{15}{16}$	$3\frac{1}{4}$	$4\frac{1}{2}$	71·1	85·0	69·6
$\frac{9}{16}$	1	$3\frac{3}{8}$	$4\frac{5}{8}$	70·0	83·0	68·0
$\frac{9}{16}$	$1\frac{1}{16}$	$3\frac{5}{8}$	$4\frac{7}{8}$	70·7	87·0	70·7
$\frac{5}{8}$	$1\frac{1}{16}$	$3\frac{5}{8}$	$4\frac{7}{8}$	70·7	78·3	64·1
$\frac{5}{8}$	$1\frac{1}{8}$	$3\frac{3}{4}$	5	70·0	84·8	69·5
$\frac{11}{16}$	$1\frac{3}{16}$	4	$5\frac{1}{2}$	70·3	80·5	66·0
$\frac{3}{4}$	$1\frac{3}{8}$	$4\frac{1}{2}$	$6\frac{1}{4}$	70·1	79·2	64·9
$\frac{13}{16}$	$1\frac{7}{16}$	$4\frac{3}{4}$	$6\frac{5}{8}$	70·0	84·1	69·0
$\frac{7}{8}$	$1\frac{1}{2}$	5	7	70·0	80·8	66·2

It will be seen by the above, that when the plates exceed $\frac{15}{16}$, sufficient area of the rivets cannot be obtained without an excessive diameter.

(3) *Lap joint and treble-riveted.*—To obtain sufficient rivet area with a lap joint when the plates are thick, three rows of rivets are necessary; the rivets are, in this case, also sometimes arranged on the *chain* plan, but more generally they are zigzag. A strength* of joint of 72 per cent. can generally be obtained with a rivet area equal to that of the plate between the holes; hence

Pitch of rivets = 3·57 × diameter.

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* By arranging the rivets so that there is half the number in each edge row that there is in the middle row, the strength of joint may be as much as 85 per cent.

Also, $$3\frac{\pi d^2}{4} = \frac{72}{100} p \times t \times \frac{28}{23} = 0\cdot876\, p \times t$$

whence substituting the above value of p,

Diameter of rivet = 1·33 × thickness of plate.

Example.—To find the pitch and diameter of the rivets for a treble-riveted lap joint, the plates being 1 inch thick.

Diameter of rivets = 1 × 1·33, or 1·33 inches.
Pitch of rivets = 3·57 × 1·33, or 4¾ inches.

The following table gives the pitch, &c., as found in general practice :—

TABLE XLII.—Lap Joints, Treble-Riveted (Zigzag).

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.	Breadth of Lap.	Percentage of		
				Plate.	Rivets.	B.T. Joint.
¾	1	3⅝	6¾	72·4	86·6	71·0
13/16	1⅛	4⅛	7⅝	72·7	89·0	72·7
⅞	1 3/16	4¼	8	72·0	89·3	72·0
15/16	1¼	4½	8½	72·2	87·2	71·5
1	1⅜	5	9¼	72·5	89·1	72·5
1 1/16	1 7/16	5¼	9⅝	72·6	87·2	71·5
1⅛	1½	5⅜	9⅞	72·1	87·6	71·8
1 3/16	1½	5⅜	10	72·1	83·0	68·1
1¼	1⅝	5¾	10¼	71·7	86·8	71·1
1 5/16	1⅝	5¾	10½	71·7	82·6	67·8
1⅜	1¾	6¼	11¼	72·0	84·0	68·8
1½	1¾	6¼	11½	72·0	77·0	63·1

The chief difficulty with lap joints is in the working of the corners of the plates where the next strake of plating covers the lap; these corners are hammered to a taper, so as to nearly conform to a circle, and the covering plate is slightly joggled, so as to lie evenly on the deformation caused by the lapping. Some boilermakers, to ensure a good fit, go to the expense of planing or milling the corners fair, and even to extend the taper part beyond the butt end of the plate itself, so as to cause as little deformation as possible.

(4) *Butt joints with double straps and single-riveted.*—This form of joint is not often resorted to, as there are two rows of rivets, and only a shearing area of twice that of the one row of rivets, besides all the expense of double straps, which entail the caulking of four seams; the sole advantage it possesses over the double-riveted lap joint is the absence of smithed corners, and that the plates lie wholly in the circle without deformation; this, however, does not compensate for the extra expense and the liability of leakage from the two extra seams.

The strength of this joint is seldom more than 65 per cent. of the solid plate, as more cannot be obtained without placing the rivets so far apart as to prevent the straps from being caulked tight. If the straps are made of *the same thickness as the plate itself*, 70 per cent. may be obtained. Taking 70 per cent. as the strength of joint, the diameter and pitch of the rivets are the same as given for the double-riveted joint, and the breadth of the lap is six times the diameter of the rivets. For example, if the plate is $\frac{3}{4}$-inch thick, the rivets should be $1\frac{5}{16}$ inches diameter, and $4\frac{3}{4}$ inches pitch, the breadth of lap being $6 \times 1\frac{5}{16}$, or $7\frac{7}{8}$ inches.

(5) *Butt-joints with double straps and double-riveted.*—This is a very general and deservedly favourite form of joint for thick plates, and when well made gives every satisfaction. There is no necessity for smithing or machining the plates, nor of joggling the covering plate of the next strake, although some boiler-makers, to avoid the caulking of the ends of the strap where it butts against the next strake, thin down the end and notch out the covering plate, so as to lap over the strap. This makes a very good joint, but is somewhat expensive, and if the plates are properly fitted, there should be no need of such an elaboration.

For each portion of plate between the holes, there is a rivet area equal to four times the area of section of one rivet; but the Board of Trade used to allow only $3\frac{1}{2}$ times the area of one rivet, and practically do that now.

Under these circumstances the following rules hold good for a joint equal to 75 per cent. of solid plate :—

$$\frac{\text{Pitch of rivets}}{\text{diameter}} = \frac{100}{100 - 75},$$

or,

$$\text{Pitch of rivets} = 4 \times \text{diameter.}$$

Also, since

$$3\tfrac{1}{2} \times \frac{\pi d^2}{4} = \frac{75}{100} p \times t.$$

$$\text{Diameter of rivets} = 1{\cdot}1 \times \text{thickness of plate.}$$

The thickness of each butt-strap must be at least $\frac{5}{8}$ that of the plate, and when the strap between contiguous strakes is simply butted against them, it is better to be of the same thickness as the plate.

Example.—To find the diameter, pitch of rivets, and thickness of straps for a butt-joint double-riveted, and equal to 75 per cent. of solid plate, whose thickness is 1 inch.

$$\text{Diameter of rivets} = 1 \times 1{\cdot}1, \text{ or } 1{\cdot}1 \text{ inches.}$$
$$\text{Pitch of rivets} = 4 \times 1{\cdot}1, \text{ or } 4{\cdot}4 \text{ inches.}$$

Thickness of long strap $\frac{5}{8}$ inch, and of short strap 1 inch if butted, and $\frac{5}{8}$ inch if fitted under the covering plates.

The following table gives the pitch, &c., as found in general practice :—

TABLE XLIII.—Butt-Joints, Double Straps, Double-Riveted (Zigzag).

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.	Breadth of Straps.	Thickness of Straps.	Percentage of		
					Plate.	Rivets.	B.T. Joint.
$\frac{3}{4}$ inch,	$\frac{7}{8}$ inch,	$3\frac{1}{2}$ inches.	$8\frac{3}{4}$ inches,	$\frac{1}{2}$ inch,	75	91·6	75·0
$\frac{13}{16}$,,	$\frac{15}{16}$,,	$3\frac{3}{4}$,,	$9\frac{3}{8}$,,	$\frac{1}{2}$,,	75	90·6	74·3
$\frac{7}{8}$,,	1 ,,	4 ,,	10 ,,	$\frac{9}{16}$,,	75	89·7	73·8
$\frac{15}{16}$,,	$1\frac{7}{16}$,,	$4\frac{1}{4}$,,	$10\frac{5}{8}$,,	$\frac{5}{8}$,,	75	88·9	73·0
1 ,,	$1\frac{1}{8}$,,	$4\frac{1}{2}$,,	$11\frac{1}{4}$,,	$\frac{5}{8}$,	75	88·4	72·8
$1\frac{1}{16}$,,	$1\frac{3}{16}$,,	$4\frac{3}{4}$,,	$11\frac{7}{8}$,,	$\frac{11}{16}$,,	75	88·0	72·3
$1\frac{1}{8}$,,	$1\frac{1}{4}$,,	5 ,,	$12\frac{1}{2}$,,	$\frac{3}{4}$,,	75	87·3	71·5
$1\frac{3}{16}$,,	$1\frac{3}{16}$,,	$5\frac{1}{4}$,,	$13\frac{1}{8}$,,	$\frac{3}{4}$,,	75	86·7	71·1
$1\frac{1}{4}$,,	$1\frac{3}{8}$,,	$5\frac{3}{8}$,,	$13\frac{1}{2}$,,	$\frac{13}{16}$,,	74	88·4	72·8

(6) *Butt-joints with double straps treble-riveted.*—This form has become a necessity, since very thick plates have been used for boiler-shells, in order to get an adequate sectional area of rivet; it also admits for the same reason of thinner plates being used when desired, so that when there are only a few butt-joints it is really more economical to adopt this form of joint. In this case, for each portion of plate between the holes, there is a rivet area equal to six times the area of section of one rivet; but, as before, the Board of Trade only allowed this as 5¼. The percentage of plate between the rivets is generally 80 per cent. with this type of joint; hence

$$\frac{\text{Pitch of rivets}}{\text{diameter}} = \frac{100}{100 - 80},$$

or,

$$\text{Pitch of rivets} = 5 \times \text{diameter}.$$

Also, since

$$5\tfrac{1}{4} \times \frac{\pi d^2}{4} = \frac{80}{100} p \times t.$$

In practice, the rivets are never less in diameter than the thickness of the plate.

The following table gives the diameter, pitch, &c., as found in general practice with steel plates and rivets; the percentage of rivet area is large, but only as required to provide for 23 tons ultimate strength against 28 tons for the plate :—

TABLE XLIV.—BUTT-JOINTS, DOUBLE STRAPS, TREBLE-RIVETED (ZIGZAG).

Thickness of Plate	Diameter of Rivet.	Pitch of Rivets.	Breadth of Straps.	Thickness of Straps.	Percentage of		
					Plate.	Rivets.	B.T. Joint.
1 inch,	1 3/16 inch,	6 inches,	1 ft. 6¼ ins.	¾ inch,	80·2	96·9	79·4
1 1/16 ,,	1¼ ,,	6¼ ,,	1 ,, 7 ,,	13/16 ,,	80·0	97·0	79·5
1⅛ ,,	1 5/16 ,,	6 5/16 ,,	1 ,, 7¾ ,,	⅞ ,,	79·2	99·0	79·2
1 3/16 ,,	1 5/16 ,,	6⅜ ,,	1 ,, 8 ,,	15/16 ,,	79·4	93·8	77·0
1¼ ,,	1⅜ ,,	6⅝ ,,	1 ,, 9 ,,	15/16 ,,	79·2	94·0	77·0
1 5/16 ,,	1 7/16 ,,	6⅞ ,,	1 ,, 9¾ ,,	1 ,,	79·1	94·3	77·3
1⅜ ,,	1½ ,,	7¼ ,,	1 ,, 10½ ,,	1 1/16 ,,	79·3	93·0	76·2
1 7/16 ,,	1 9/16 ,,	7½ ,,	1 ,, 11¼ ,,	1⅛ ,,	79·1	93·3	76·3
1½ ,,	1⅝ ,,	7¾ ,,	2 ,, 0¼ ,,	1⅛ ,,	79·0	93·6	76·7

An improved form of treble-riveted joint is made by omitting alternate rivets in the outer rows. In this case, there are five rivets for each space in the outer row, and, allowing only 1¾ times for the double shear, there will be 1¾ × 5 times the area of one rivet for each space or 8¾. In this way a larger percentage of joint can be obtained, amounting in practice to about 84; hence

$$\frac{\text{Pitch of rivets in outer row}}{\text{diameter}} = \frac{100}{100 - 84}$$

$$\text{Pitch of rivets} = 6{\cdot}25 \times \text{diameter}.$$

Also, since
$$8\tfrac{3}{4} \times \frac{\pi d^2}{4} = \frac{84}{100} p \times t.$$

$$\text{Diameter of rivet} = 0{\cdot}77 \times t \text{ (if of iron)};$$
$$\text{,,} \quad \text{,,} \quad = 0{\cdot}96 \times t \text{ (if of steel)}.$$

In practice, the rivets in this class of joint are of the same diameter as the thickness of plate.

The following table gives diameter, pitch, &c., as found in general practice with steel plates and rivets:—

TABLE XLV.—BUTT-JOINTS, DOUBLE STRAPS, DOUBLE-RIVETED WITH EVERY ALTERNATE RIVET OMITTED IN OUTER ROW (ZIGZAG).

Thickness of Plate.	Diameter of Rivet.	Pitch of Rivets. Outer Row.	Breadth of Straps.	Thickness of Straps.	Percentage of		
					Plate.	Rivets.	B.T. Joint.
1 inch,	1 inch,	6⅜ inches,	1 ft. 3½ ins.	13/16 inch,	84·3	107	84·3
1 1/16 ,,	1 1/16 ,,	6¾ ,,	1 ,, 4 ,,	⅞ ,,	84·2	108	84·2
1⅛ ,,	1⅛ ,,	7⅛ ,,	1 ,, 4½ ,,	⅞ ,,	84·2	108·5	84·2
1 3/16 ,,	1 3/16 ,,	7⅝ ,,	1 ,, 5¾ ,,	15/16 ,,	84·4	110	84·4
1¼ ,,	1¼ ,,	7⅝ ,,	1 ,, 6¼ ,,	1 ,,	83·6	112	83·6
1 5/16 ,,	1 5/16 ,,	7⅝ ,,	1 ,, 6½ ,,	1 1/16 ,,	82·78	111	82·8
1⅜ ,,	1⅜ ,,	8 ,,	1 ,, 7½ ,,	1⅛ ,,	82·8	118	82·8
1 7/16 ,,	1 7/16 ,,	8 ,,	1 ,, 8¼ ,,	1 3/16 ,,	82	109	82·0
1½ ,,	1½ ,,	8½ ,,	1 ,, 9 ,,	1¼ ,,	82·3	121	82·3

The new rules of the Board of Trade for riveting are given in the Appendix.

(7) *Welded joints.*—To avoid all chances of a leakage from the longitudinal joints, some boiler-makers have welded the butts of the plates together. The Board of Trade allowed only 75 per cent. of solid plate as the value of this joint, and have now withdrawn their consent to this method altogether. Although it is quite possible to make a perfect weld, and it is often done in boiler-making, there is always some degree of uncertainty in the process, and no one can have confidence in what cannot be proved to be right. The goodness of this joint depends on the skill and care exercised by the workmen; and, as an imperfection is easily covered and concealed, no one with a due sense of responsibility could accept such joints, unless made wholly under his personal inspection, and even then there is not that sense of security which is felt with a riveted joint.

Circumferential Seams are almost always lapped and double-riveted. Boilers of small size and for comparatively low-working pressures are single-riveted, and as one row of rivets is quite sufficient to ensure tightness of joint, which is all that is required in this joint, the single-riveted joint might be adopted more generally than now obtains in single-ended boilers. Boilers made for very high pressures are often treble-riveted in the circumferential seams, the Board of Trade rules admitting of a thinner plate when this is done. In the case of large double-ended boilers, the circumferential seams should be always treble-riveted except the end ones, which need be done in this way only when the plates are very thick.

Methods of Work.—Formerly the holes in boiler plates were punched and drifted fair. Since the Board of Trade placed a premium on drilled holes, machines have been made which compete successfully, both in the cost and speed of the work, with the punching machine, and all the holes in a boiler shell are drilled *in place* at a trifling expense, if any, beyond that of punching.

Material.—The shell plating of a cylindrical boiler, when made of iron, was usually of the quality known as "Best Staffordshire"; that is, the iron is supposed to have a tensile strength with the grain of about 48,000 pounds per square inch, and to be equal in all other respects to Staffordshire boiler plates branded as "Best." In selecting plates for a cylindrical shell, it is necessary to bear in mind that tensile strength and moderate elasticity are the two qualities requisite; the Board of Trade, Lloyd's, and the British Corporation permit the use of steel plates with a strength of 32 tons, while the Admiralty limit it to 30 tons; it is expected that an extension of 20 per cent. is made before fracture (see Table xxxix.). Steel of a higher strength is sometimes used, and it is likely that the above authorities will shortly permit of the general use of it, with 35 tons as the limit and with 18 to 20 per cent. extension.

Siemens steel has now taken the place of iron in boiler-making, and, as it possesses a much higher tensile strength, together with greater elasticity, it is a more suitable material for the purpose. A boiler made wholly of steel is also cheaper than one made wholly of iron, which no doubt was one cause of the increased demand for steel boilers. Steel plates can, at a trifling extra expense, be supplied of very large sizes, exceeding those made by the Yorkshire ironmasters in magnitude. As a matter of fact, the overhead price of a heavy specification of steel plates is generally very nearly the same as that of a light specification, while there is a very considerable difference for iron if large and heavy plates are included. The size of plates to be used in the construction of the shell depends now on the appliances of the boiler-maker; the breadth of plate is limited by the depth of gap in the riveting machine, and the length of plate by the capabilities of the planing machine and squeezer or rolls; and the weight of the plate is limited to the strength of the various small cranes, &c. Steelmakers can make plates up to 12 feet broad and almost of any length.

Nowadays rolls or squeezers for curving plates, planing machines for truing the edges, drilling machines for dealing with all the holes, and riveting machines for closing the joints are made in such large sizes that a double-ended boiler of the largest size can be made in three drums, and each drum in two plates; single-ended boilers in one drum, or two if very long, and each drum in two plates; some people prefer to make each drum in one plate when the diameter permits of it. As a rule, it may be assumed that a single-ended boiler has one or two drums or strakes of plating, each strake consisting of two plates; a double-ended boiler should have three strakes of plating: the longitudinal seams should be so arranged that no two of them come in line nor interfere with the seams at the ends, and they should be well above the furnace line.

Allowance for Wear.—All boilers are designed so as to last as long as possible, and, since wear takes place by corrosion, some additional thickness must be provided at first to meet this condition. The Board of Trade tacitly make this allowance by using a high factor of safety; but, since the factor of safety causes the additional thickness to be *proportional* *to the total thickness*, while the wear takes place independently of thickness, it does not properly meet the case. A boiler with plates $\frac{1}{2}$ inch thick will waste the same quantity of iron per square foot as one with plates 1 inch thick if worked under similar conditions. Suppose such waste to be $\frac{1}{8}$ inch in a certain time, the loss in one case is 25 per cent., but only $12\frac{1}{2}$ per cent. in the other. To meet the case properly, the factor of safety should be reduced, and a constant quantity added; for example, in the case mentioned above the plates should be $\frac{9}{16}$ inch and 1 inch, so that at the end of the time the former should be $12\frac{1}{2}$ per cent. under $\frac{1}{2}$ inch in thickness. If any further proof be needed, it is only necessary to calculate the thickness of plates for boilers of small diameter and low pressure to find it such as would be impossible to rivet and caulk tight.

The following rules make due allowance for such contingencies:—Let D be the diameter of the shell in inches, p the working pressure in pounds per square inch, F a factor; then

$$\text{Thickness of shell plate} = \frac{D \times p}{F} + \frac{1}{8} \text{ inch.}$$

The following are the values of F:—

Description of Joint.	Mild Steel.
Lap joints, single-riveted, - - - - -	16,000
,, ,, double- ,, - - - - -	21,500
,, ,, treble- ,, - - - - -	22,000
Butt ,, double- ,, double straps, - -	22,500
,, ,, treble- ,, ,, ,, - -	26,000

Example I.—To find the thickness of plates for a boiler shell 8 feet 4 inches diameter for a working pressure of 60 lbs., lap joints single-riveted.

$$\text{Thickness} = \frac{100 \times 60}{16{,}000} + \frac{1}{8} \text{ inch} = \frac{1}{2} \text{ inch.}$$

Example II.—To find the thickness of plates for a boiler shell 15 feet diameter for a working pressure of 160 lbs., butt straps treble-riveted.

$$\text{Thickness} = \frac{180 \times 160}{26{,}000} + \frac{1}{8} \text{ inch} = 1\frac{1}{4} \text{ inch.}$$

Boiler Ends.—There are several methods of connecting the end-plates to the shell.

(1) *By means of angle irons.*—This is the oldest method, and copied from the practice of makers of land boilers. It was the cheapest when there were no special appliances for flanging, and also because it admitted of the end-plates being of an inferior quality of iron compared with what is necessary when required to be flanged; but there are double the number of rows of rivets and caulking edges from which leakage may take place, and the angle iron is not well calculated to withstand the strains. From a constructive point of view, this plan has special advantages, for when the angle iron is external (fig. 157) there is more room on the boiler front for the furnace mouths, man-holes, mountings, &c.; and with the angle iron the lapping of the end-plates does not tend to cause leakage at the corners and

altogether the work on it is of a simpler nature. For single-ended boilers, the back angle-iron internal (fig. 158) and the front external (fig. 157) made a very good arrangement.

(2) *Flanging the end-plates.*—This is now the universal and, on the whole, the best plan, for there are only one set of rivets and two caulking edges, and the room occupied is less than with the external angles. Flanging is done by special appliances at a trifling cost in various designs of hydraulic press. The flanges (fig. 161) are usually inside the boiler, but when it is of so small a size that the rivets cannot be held up inside, or it is desired to rivet the joint by machine, the *front* end is placed (fig. 160) with the flange outside. In either way there is no tendency to force the joint by pressure on the ends, as there is with angle irons.

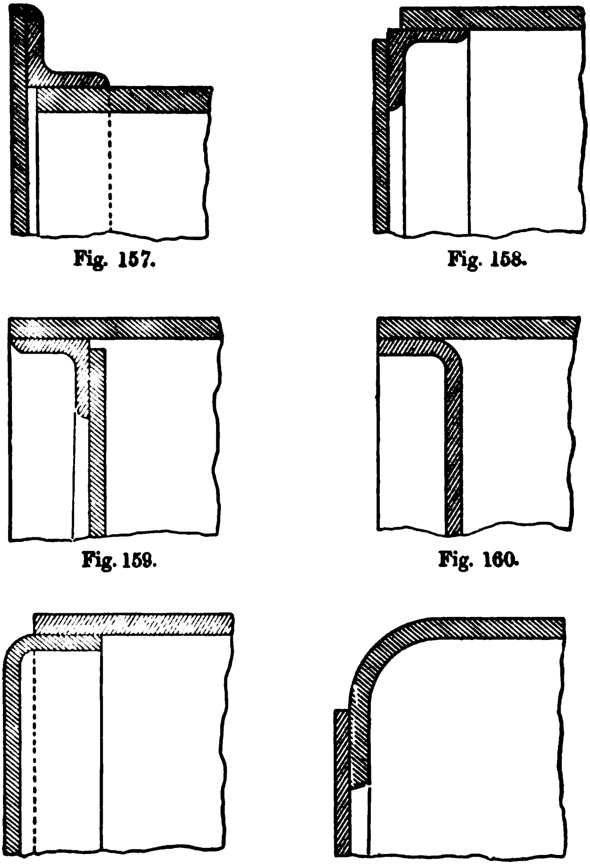

Fig. 157. Fig. 158.

Fig. 159. Fig. 160.

Fig. 161. Fig. 162.

Figs. 157 to 162.—Boiler Ends.

(3) *Flanging the shell plates.*—Some few engineers, in order that a thicker plate may be provided to withstand the wear that takes place at the

bottom corners, or rather edges, of the cylindrical boiler, and to avoid the fitting of the flanged ends into the shell with fair accuracy, flange the shell plates (fig. 162) inwards, so as to form a connection for the ends. This has some bad features, among which may be cited the difficulty and cost of flanging thick plates across the grain, especially after being bent; the necessity for such shell plates to be of low tenacity to have so much ductility; and lastly, the pressure on the end always tending to open the joints. It has, however, some of the advantages of the angle irons, but this advantage is dearly bought, for, to avoid the chance of leakage at the corners of the end-plates, there is far greater risk of leakage from the longitudinal shell joints, which are of necessity flanged over too, unless welded. Also, such an arrangement quite prevents the furnaces from being near the shell.

Riveting.—The cross seams of the backs are usually double-riveted, although for moderate pressure single-riveting does quite well. The riveting of the ends to the shell is usually of the same design as that of the other circumferential seams, but, in double-ended boilers, they need not be treble-riveted because the middle seams are.

The Quality of Plate depends largely on the amount of flanging to be done. To stand flanging around the edges of the circular plates, they may be of the same quality as the shell; but, when the furnace holes are flanged to meet the furnaces, a softer quality is necessary, and it is better to use the same as ordered for internal parts, with an extension of 25 per cent. at least to stand such severe treatment.

The Thickness of the End-plates and the pitch of the stays are interdependent to a certain extent; but, since the stays in the upper part of the boiler must be wide enough apart to admit of a man passing between them, the plates at the upper part of the ends must be made thick enough to suit this pitch of the stays, which are usually, in consequence, somewhat thicker than the other part of the ends; some makers, however, prefer to make the ends of one uniform thickness, and stiffen the top plates to stand the wide pitch of the stays by riveting on thick and large washers in wake of these stays. The end-plates are generally from $\frac{5}{8}$ to $\frac{7}{8}$ inch thick, although a few makers prefer to have thicker plates, so as to avoid the necessity for doubling plates about the man- and mud-holes, and for fitting nuts and washers to the screwed stays. Taking 15 inches as the smallest convenient pitch for the stays in the steam space, and suppose them to have riveted washers as well as double nuts, Lloyd's requires the plate to be $\frac{13}{16}$ inch thick for 150 lbs. working pressure, and $\frac{15}{16}$ inch thick for 200 lbs. The Board of Trade permit $\frac{1}{32}$ inch less for these pressures.

Furnaces.—Those fitted in the cylindrical boiler are invariably of circular section, that being the best form to resist a uniform external pressure. The strength of such a furnace varies as the square of the thickness of plate, and inversely as the diameter and length. From experiments made on a large scale, there is, however, reason to doubt the supposition that the strength varies *exactly* inversely as the length.

Let D be the external diameter, in inches, of the furnace whose length is L feet, t the thickness of the plates in parts of an inch.

$$\text{Safe working pressure} = \frac{99{,}000 \times t^2}{(L+1) \times D} \quad \text{(Board of Trade).}$$

Or,

$$\text{Safe working pressure} = \frac{89{,}600 \times t^2}{L \times D} \quad \text{(Lloyd's Registry).}$$

When the plain part of a furnace exceeds 120 times the thickness of the plate,

$$\text{Safe working pressure} = \frac{51{\cdot}5}{D}(18{\cdot}75t - 12{\cdot}36L) \quad \text{(British Corporation).}$$

If the furnaces are constructed by welding the plates together, or connecting them by a butt joint with double straps single-riveted, or by a single butt strap double-riveted, the Board of Trade allow the above factor. To avoid making from the above rules a furnace which might give way by the crushing of the material, the Board of Trade insist that, in no case shall the working pressure, in pounds per square inch, exceed $\frac{9900 \times t}{D}$, and Lloyd's that the working pressure shall not exceed $50\left(\frac{300t - 12L}{D}\right)$, when the length of the plain part is less than 120 times the thickness of plate.

The following rule gives a suitable thickness for the furnace plates in 32nds of an inch, and makes *due allowance for uniform wear of the surfaces* :—

$$t = \sqrt{\frac{p \times L \times D}{F}} + 2.$$

Here L is the length and D the diameter, *both in inches*; and F a factor which is for ordinary mild steel, 1000.

Furnace plates are made from $\frac{3}{8}$ inch to $\frac{3}{4}$ inch thick, the most general sizes being $\frac{7}{16}$ to $\frac{9}{16}$ inch. Some engineers, however, in spite of this, to avoid using corrugated furnaces or fitting stiffening rings, make the furnace of plates as thick as $\frac{7}{8}$ inch; if the boiler is kept quite clean this will do.

If the furnace is so long that $\frac{9}{16}$-inch plates are insufficient by the rules for the working pressure, some means of stiffening it should be resorted to; such stiffening was generally effected by means of rings secured to the plates, and as the effective length for purposes of calculation is thus virtually the longest distance between such rings, or between such rings and the ends, it may be taken as the value of L in the foregoing formulæ.

The Methods of Stiffening Furnaces were—(1) *By means of a hoop* (fig. 163) formed of two flat bar rings riveted together, parallel with one another, and at a sufficient distance apart to admit of a stud passing between them. The furnace is held to this hoop by means of studs screwed into it, and, passing between the rings, fitted with nuts and washers. This is a cheap plan, and a very convenient one for stiffening an existing furnace; as is also the following:—

(2) *By means of a hoop* (fig. 164) formed of two angle-iron rings riveted together; the furnace is held to it by studs screwed into it, and, passing either between the angles or through the flanges, fitted with nuts, &c.; when passing through the flanges, it is usual to fit a thimble to each stud.

(3) *By making the furnace in two or more drums, and connecting them by means of a* U*-shaped hoop*, called the "Bowling hoop" (fig. 165), because first made by the Bowling Iron Company. These hoops are weldless, and possess a considerable amount of elasticity; so that, in addition to stiffening, they allow expansion longitudinally on the part of the furnace. This was a very convenient plan, as it admitted of the furnace being partially withdrawn in case of damage, &c., and notwithstanding that there are two thicknesses of plates, and two laps at each joint of the furnace, it gave every satisfaction when tried. No one makes these hoops now, so that this type of joint has gone out of use.

(4) *By making the furnace in two or more drums, and connecting them by means of flanges*, formed by turning the plate end outwards (fig. 166). To allow of a caulking edge on both sides of the lap, a thin ring is introduced

between the flanges. This was a favourite method, because no joint or riveting is exposed to the fire; but it is expensive, and requires care both in the flanging and in the fitting together to avoid trouble. Such joints generally give trouble from the strain on the boiler end tending to open the joint, and by the wearing away of the metal at the root of the flanges, due to mechanical action. Furnaces made on this plan also require more room in the boiler, and the flanges block up the space between them and the boiler shell.

(5) *By making the furnace in two or more drums of different diameters;* those of small diameter are flanged out so as to fit into, and be connected to, the larger ones with lap joints single-riveted (fig. 167). This is known as Hawksley's patent; and, although successfully applied to land boilers, has, so far, only been tried on a small scale in marine boilers. It possesses one or two very useful features, among which may be reckoned the capability of the furnace, being small at the mouth, to leave good space on the boiler fronts for man-holes, &c., and while small at the combustion-chamber ends, it is of good diameter in the middle. It has, however, the objection of presenting joints to the direct action of the fire.

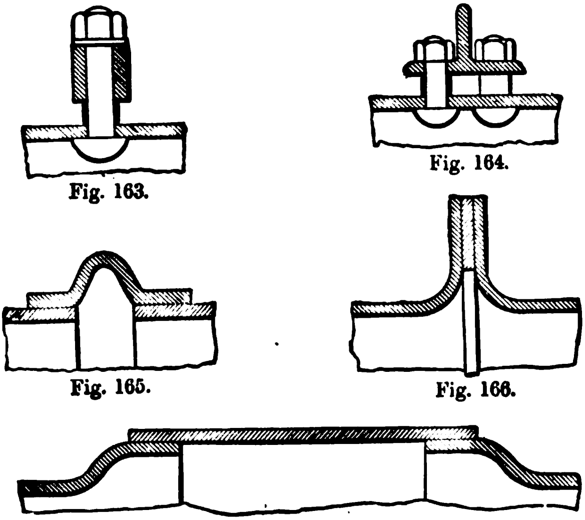

Fig. 163.

Fig. 164.

Fig. 165.

Fig. 166.

Fig. 167.

Figs. 163 to 167.—Methods of Stiffening Furnaces.

(6) *By making the furnace with a series of corrugations or ridges.*—There are now several ways of accomplishing this, the best known being that of Mr. Fox (fig. 169), as made by the Leeds Forge Company, and that of Mr. Purves (fig. 170) made by John Brown & Co. of Sheffield. A more recent patent is that of Mr. Morison (fig. 171), and made by the Leeds Forge Company; this form possesses the good features of the Purves, and avoids the fault of the Fox. Fig. 172 is the form of furnace made by C. D. Holmes & Co., Hull. Fig. 173 is the form in which Mr. Deighton, of Leeds, makes

furnaces which have proved most satisfactory; he also makes the ordinary Fox corrugated furnace.

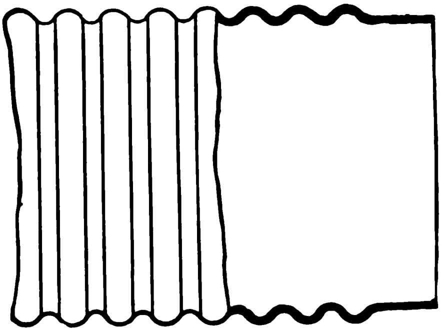

Fig. 168.—Fox's Furnace.

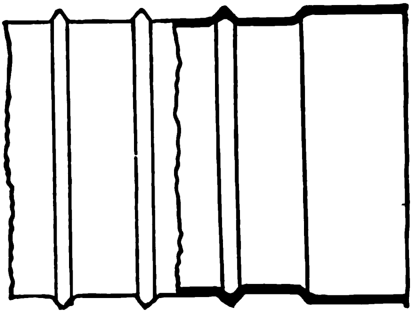

Fig. 169.—Purves' Furnace.

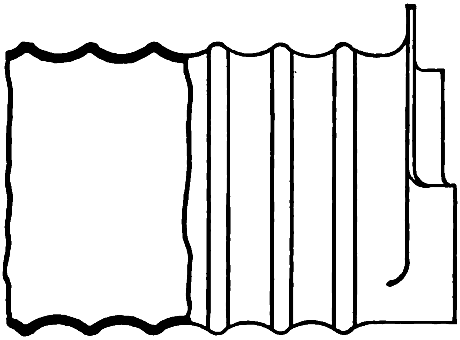

Fig. 170.—Morison's Furnace.

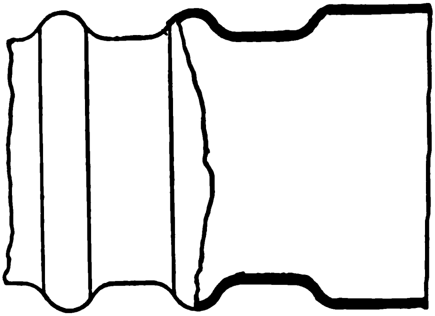

Fig. 171.—Holmes' Furnace.

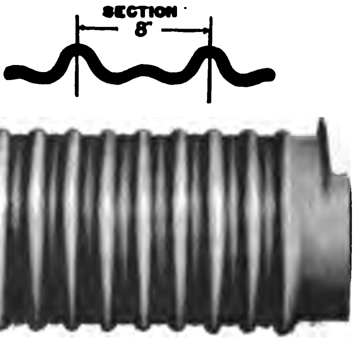

Fig. 172.—Deighton's Furnace.

The corrugated furnace is an extension of the Bowling-hoop principle, and its genesis is best illustrated by reference to fig. 171, which shows the plan followed by Mr. Holmes. Here there are comparatively few corrugations, but still sufficient to give the necessary stiffness to the furnace. For increased pressure, such a furnace as this must either be made of thicker plate or have the corrugations closer together; consequently, for the same pressure and dimensions, the Fox furnace will be thinner than that of Mr. Holmes. On the other hand, the Holmes furnace is more rigid longitudinally than the Fox furnace. The corrugated furnace made by the Farnley Company has the corrugations formed spirally around the furnace, and are said thereby to give greater longitudinal rigidity, without sacrificing too much of its transverse stiffness; there must, however, be set up twisting strains when end-pressure is applied, which would tend to shear the rivets, and the transverse stiffness cannot be any more than that of the Holmes furnace.

These furnaces have become a necessity for large diameters and high pressures; but, although immensely strong so long as the metal is cold, corrugated ones will probably collapse *longitudinally* when red hot quicker than an ordinary furnace, from the fact of there being superabundance of plate between the extreme points of support to supply the extra length of the arc over that of the chord; a common furnace cannot come down in this way without stretching the metal; in the Fox furnace the corrugations are simply drawn out of shape.

The Purves furnace is an extension of the Adamson-joint principle, and is shown in fig. 169. This furnace is said to possess quite as much transverse stiffness as the Fox, while possessing superior longitudinal rigidity; it is, moreover, easier to clean and to repair.

For corrugated furnaces, the following rules hold good:—

(*a*) $$\text{Working pressure} = \frac{14{,}000 \times \text{thickness in inches}}{D} \text{ (Board of Trade),}$$

D being taken as the smallest outside diameter in inches.

(*b*) $$\text{Working pressure} = \frac{C \times (T - 2)}{D} \text{ (Lloyd's Register).}$$

T is the thickness in sixteenths of an inch.
D is the smallest diameter outside.
C = 1259 for Fox's, Morison's, Deighton's, Beardmore's, or Leeds Forge bulb furnaces where made of steel, 26 to 30 tons tensile strength.
C = 1160 for Purves', when the ribs are 9 inches apart; or Brown's, ribs 8 or 9 inches apart.
C = 945 for Holmes', and C = 912 for Farnley's.

(*c*) $$\text{Working pressure} = \frac{C \times (T - 2)}{D} \text{ (British Corporation).}$$

C = 1250 for Leeds Forge bulb furnaces.
C = 1160 for Fox's, Purves', Deighton's, Morison's, and Brown's.
C = 950 for Holmes' and Farnley's.

(*d*) $$\text{Working pressure} = C\,\frac{(T - 2)}{D} \text{ (Bureau Veritas);}$$

C = 1259 for corrugated and bulb furnaces.
C = 1160 for ribbed furnaces.

Fig. 173 shows Morison's furnace with its back end so formed as to permit of the furnace being withdrawn through the hole in front of the boiler, without sacrificing any of the good features of the method of jointing of the combustion chamber.

Messrs. John Brown & Co. have also a furnace with a back end formed so as to permit of the furnace being withdrawn in the same way. In this case, however, the furnace is reduced somewhat in diameter and flanged outward all round. This flange is not concentric with the body of the furnace, its centre being higher so as to permit of the connection of the tube plate in the usual way, the back tube plate being flanged and riveted into the combustion chamber all round, and formed with a round hole corresponding to that of the furnace neck.

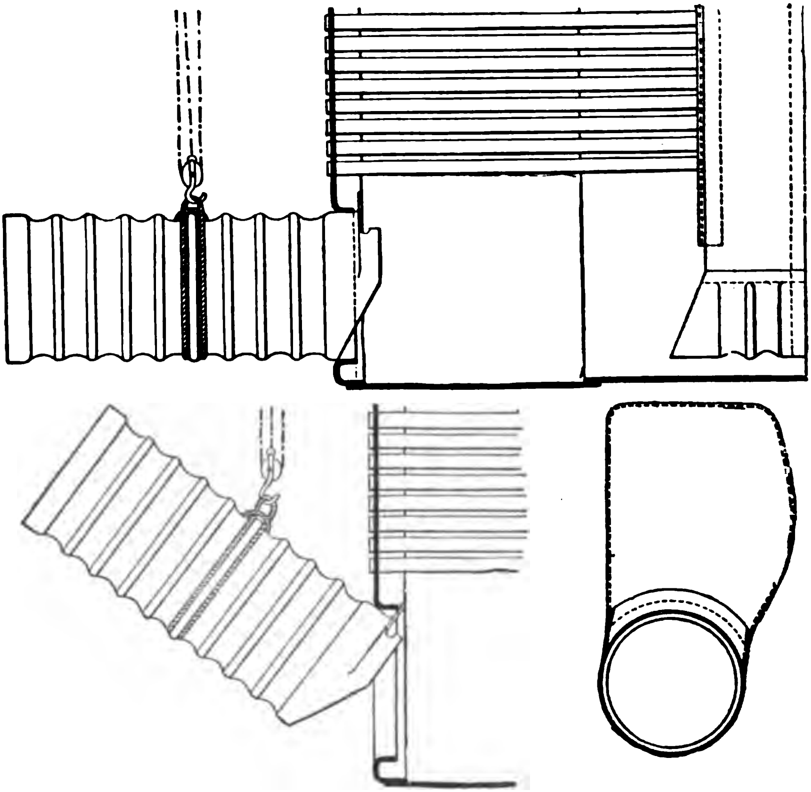

Fig. 173.—Removable Furnace by the Leeds Forge (Ashlin's Patent).

Methods of Connecting Furnaces to End-plates.—There are three distinct ways of accomplishing this:—

(1) *By means of angle-iron rings* (fig. 174) outside of the furnace, and either inside or outside of the boiler end, and in some rare cases inside the furnace and inside the end. This is the older method, and copied from the practice of the land boiler-makers. It was a very cheap method, and a convenient one when expert smiths could not be obtained, or when the plates were not soft enough to stand flanging; but there is always an objection to strain angle-iron across the grain, and two rows of rivets and four caulking edges are never so good as half that number.

(2) *By flanging the furnace to meet the front* (fig. 175).—This is a great improvement on the angle-irons, as it avoids the two chief defects of that system, but still retains one—viz., that the pressure on the ends tends to

open the joint. The flanging is invariably outwards, and the root should have good curvature, the radius of the outer surface being at least 1½ inches. Since steel has been used, this method has been practically dropped, as it does not admit of the furnace being withdrawn in case of need.

(3) *By flanging the front plate* inwards (fig. 176) or outwards (fig. 177) to meet the furnace. The former is the most general method, the latter being resorted to only when the boiler is so small as to prevent the rivets from being properly "held up," and when it is necessary to have the furnaces very near the shell. In naval ships the boilers are very generally made in this way. This makes the best finish to the boiler front, and leaves more room for man- and mud-holes, &c., and permits of much neater furnace fronts. The joint (fig. 176) is capable of being caulked at both edges, and the strain on it is across the rivets, and does not, in consequence, tend to start the caulking.

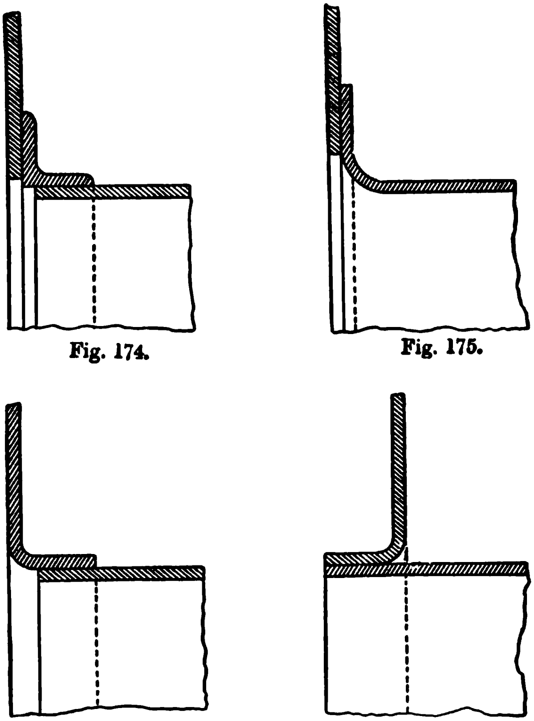

Fig. 174. Fig. 175.

Fig 176. Fig. 177.

Figs. 174 to 177.—Furnace Ends.

Longitudinal Joints of Furnaces.—When plain furnaces are fitted, the plates are better joined by welding, where there are men capable of doing the work. With proper appliances there is little difficulty in performing this operation, and, with practice, the men become very expert and sure of their work. There are two methods of effecting the weld, the one consisting in turning the edges up after slightly "upsetting" them, bringing them closely together over a strong flame till raised to a welding heat, when they are hammered down flat on a suitable anvil, placed inside the furnace; the other method is preferable when the plates to be joined are of the same quality, as it consists of inserting a bar between the two edges which have been previously "upset" and bevelled, the plates and bar being raised to the welding heat in a separate fire. Steel of special section is often used now to form the "glut" and preserve an uniform thickness. If the plates are connected by a riveted joint, there is a great advantage in welding the end of the joint, so that no leakage may occur from it where it joins the combustion chamber and boiler end.

The next best form of joint for plain furnaces is the butt with double straps and single-riveted, and this was the one adopted when steel was first employed. When connected by butt straps, the steel plates may be, with advantage, welded at the ends of the joint.

Combustion Chambers.—The length of a combustion chamber, measured in line with the furnace, should be such that its capacity above the level of the fire-bars is equal to the total capacity of the furnace, when the boiler is single-ended; when double-ended and one combustion chamber is common to opposite furnaces, the capacity of the combustion chamber should be equal to three-fourths of the combined total capacity of the two furnaces.

To obtain such a capacity of combustion chamber when the boiler is single-ended, or double-ended and divided transversely, the length must be about two-thirds the diameter of the furnace, and when common to two opposite furnaces, it must nearly equal the diameter of furnaces.

Combustion chambers are sometimes formed with flat tops (fig. 133), and sometimes by curving the back plate over the top to meet the flange of the tube plate (fig. 130). The latter plan avoids the necessity of the girder stays required to stay the flat top, and reduces the number of joints of plating, but the capacity of the combustion chamber is less, and the space for tubing, &c., contracted. It is often claimed for this form that staying is avoided, but this is not a substantial gain, as, in a single-ended boiler, the stays which are necessary for the back end-plates form the stays of the chamber, and in a double-ended boiler, if stays are omitted between the chambers, the Board of Trade surveyors require additional staying in the steam space *to tie the ends of the boiler together*. Although the plan was a favourite one with many engineers a few years ago, it is now seldom seen.

The thickness of plates and pitch of stays are of course interdependent, but, as a rule, the chambers of large boilers, whose working pressure is 150 lbs. per square inch and upwards, are made of $\frac{1}{2}$-inch to $\frac{5}{8}$-inch plates, and those of smaller boilers, or those working at lower pressures, are made of $\frac{7}{16}$-inch to $\frac{1}{2}$-inch plates.

By the Board of Trade rules, a stay-bar $1\frac{3}{8}$ inch in diameter, screwed 10 threads per inch, will sustain 73 square inches at a working pressure of 150 lbs., and only 55 square inches at 200 lbs., while a $1\frac{5}{8}$-inch stay, with 9 threads per inch, will sustain 77 square inches at a pressure of 200 lbs. A plate $\frac{1}{2}$ inch thick requires a stay for 7·74 inches pitch, or 60 square inches at a pressure of 150 lbs., while a $\frac{5}{8}$-inch plate requires one for 9·3 inches pitch or 86 square inches for that pressure; for 200 lbs. pressure, the $\frac{5}{8}$-inch plate requires a stay for a pitch of 8 inches or 66 square inches.

Board of Trade Rule:—Pressure $= \dfrac{C \times (T + 1)^2}{S - 6}$.

T is the thickness of flat plate in sixteenths of an inch; S, area of surface supported in square inches; C, a constant which, for screwed stays with nuts and plates exposed to flame, but in contact with water, is 100.

For stays in steam space fitted with riveted washers, two-thirds the pitch in diameter, nuts, &c., 210 if washers are of the same thickness as plate, but only 165 if with plain washers two-thirds the thickness of plate and three times the diameter of stay.

The bottom of the chambers should be $\frac{1}{16}$ to $\frac{1}{8}$ inch thicker than the sides, as from various causes there is often rapid wear in that part; also to avoid excessive staying and to provide for burning, which sometimes takes place there, the plates at the top should be $\frac{1}{16}$ inch thicker.

The back tube plates vary in thickness from $\frac{9}{16}$ in small boilers, and in boilers for low pressures, to $\frac{3}{4}$ inch, and even $\frac{7}{8}$ inch, in large ones for high pressures. Generally, in modern boilers of ordinary sizes and pressures, the back tube plate is $\frac{5}{8}$ to $\frac{3}{4}$ inch thick, the former being the best size when possible, as with it the tubes can be made quite tight, and there is less liability of cracking the plates or burning the tube ends than with the thicker plates.

The back plate and tube plate of the combustion chamber are almost invariably flanged inwards to take the side plates and those on the top and bottom; some makers have tried to make the chambers by flanging the sides top and bottom to meet the back and tube plates; but, as this is very troublesome to effect and prevents the tubes from being extended to the sides and top, it is seldom followed. The flange of the top plate of the furnace should be inside the chamber, and connected to the tube plate with counter-sunk rivets; the landing edge is then turned away from the "wash" of the flame, and no rivet heads are exposed to it. The landing edges of all joints of plating exposed to water should be downwards, so that deposit cannot lodge on them; when they are upwards on the water side the deposit on them is very considerable, and it is found that rapid corrosion then takes place in the angle beneath it.

Tubes.—The tubes in the ordinary marine boiler are from $2\frac{1}{2}$ inches to 4 inches external diameter, the usual sizes being from $2\frac{1}{2}$ inches to $3\frac{1}{2}$ inches in the mercantile marine, and $2\frac{1}{2}$ inches to $2\frac{3}{4}$ inches in H.M. Navy. With the ordinary natural draught the tubes should not be more than 24 diameters long; with the forced draught they may be as much as 60 diameters long, as in a locomotive boiler, but the practice in torpedo boats and steam launches is to make the tubes about 35 diameters long. The length, however, does not matter so much with forced draught.

The spacing of the tubes often depends on circumstances, but in the mercantile marine, where space and weight of machinery are not of such moment as in the Navy, the pitch of the tubes is usually 1·4 × diameter. There is less liability to prime when the tubes are widely spaced, and they are more easily cleaned from scale. For the latter purpose they are arranged in rows, both horizontally and vertically, and not zigzag, as often seen in locomotive boilers.

Tubes are manufactured of a certain minimum thickness, and said to be "according to list" when so made. If the pressure they are intended to withstand does not exceed 60 lbs., they may be "according to list"; if it does not exceed 100 lbs., they should be "1 gauge thicker than the list"; if the pressure does not exceed 150 lbs., the tubes should be "2 to 3 gauges thicker than the list."

The following table gives the thickness under the various circumstances in the numbers of the Legal Standard Gauge :—

TABLE XLVI.—Boiler Tubes.

External diameter of tubes, Ins.,	2	2¼	2½	2¾	3	3¼	3½	3¾	4
Thickness for 60 lbs., · L.S.G.,	12	12	11	11	11	10	10	10	9
Thickness to 100 lbs., · L.S.G.,	11	11	10	10	10	9	9	9	8
Thickness to 150 lbs., · L.S.G.,	10	10	9	9	8	8	7	7	6
Thickness to 200 lbs., · L.S.G.,	9	9	9	8	8	7	7	6	6

It is usual to make the tubes of slightly larger ($\frac{1}{16}$ to $\frac{1}{8}$ inch) diameter at their front end, so as to draw out easily when once started from the plates. Tube manufacturers will swell the ends to $\frac{1}{16}$ inch larger diameter without extra charge.

Boiler tubes are made of iron, steel, hard copper, and brass; when of iron they are manufactured from strips of the best Staffordshire iron, *lap* welded; when of steel they are often made lap welded like the iron ones, but more generally they are solid drawn—those of water tube boilers are always solid drawn for naval purposes; when of copper or brass they are usually solid drawn, and consequently without a joint. For ordinary marine purposes, copper and brass tubes are seldom or never used now.

Iron tubes are almost always used in the mercantile marine; brass tubes were used in the Navy, partly because of their superior conducting power, but chiefly on account of their endurance and reliability, as the iron tubes then made seldom lasted more than four years, and after only as many days a few would sometimes prove defective from small holes being formed, which rapidly enlarged with the rush of water and steam through them.

Brass tubes are made of a composition of 68 per cent. of B.S. copper and 32 per cent. of spelter, which, when drawn out into the tube, has a very high tensile strength and is very tough. Such tubes lasted ten or twelve years under ordinary circumstances, but, if used with coal containing much sulphur, they perished more rapidly and lost the toughness. They cost about four times the price of iron tubes, but, when condemned, are worth about half the original cost, and, since they lasted at least double the time and their superior efficiency was a sufficient set-off for interest on capital, the brass tubes were more economical than the iron ones. The chief objection to their continued use, however, is partly a financial one, as the prime cost is higher, and insurance, &c., depends on it as well; but the fact that, when dirty, their conducting power is very little better than iron ones, and that, with forced draught, the scour soon damages the ends near the fire, together with the much longer life of iron or steel tubes consequent on better care, have caused the abandonment in both Navy and mercantile marine of brass tubes.

Steel tubes are now being used on a large scale; but the early experience with this metal for tubes is no more happy than that of some few engineers with it for boiler plates; there is steel and steel, and because tubes of that material failed egregiously many years ago, it is no reason why tubes made of modern steel should not be used now, especially in steel boilers. The Admiralty use welded steel tubes in cylindrical boilers, and the tube-makers are assured that they can make as perfect a weld with the mild steel strips they use as with iron. For internal pressure, however, a solid-drawn tube is preferred.

To preserve the ends of the tubes in the back tube plate from being wasted away and the severe action of flame on the tuoe plates, the Admiralty direct that iron ferrules are to be fitted to them. These ferrules (fig. 178) are made of malleable cast iron; they have a slight taper and a mushroom-shaped flange, so that when driven home they are very tight and protect the end of the tube and surrounding plate completely.

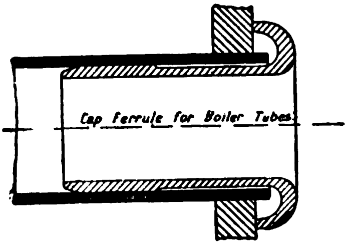

Fig. 178.

Stay Tubes.—The tube plates are usually held and stiffened by some of the tubes of greater thickness being arranged as stays; for that purpose they are screwed at the ends with a fine thread (10 to 12 threads per inch), and either tapped into both plates or tapped into the back plate and screwed by nuts to the front plate. The better plan is the former, the back end thread is *minus* and the front end *plus*, and the tube screwed into both plates at the same time. The section of thread in this plan is in excess of that of the tube owing to its large diameter, and the tube can be withdrawn at any time without disturbing the others. If fitted with nuts, there is great difficulty in getting the tube out, generally necessitating the withdrawal of a whole row, while there is really no difficulty in tapping the holes, and no necessity for nuts. The stay tubes in the Navy have a *plus* thread at each end, the diameter of the front end being larger than that at the back; in this case, the thickness at bottom of thread is the same as that of the body of the tube.

Stay tubes are usually $\frac{1}{4}$ inch thick in the body, and as the thread is $\frac{1}{16}$ inch deep, it is only $\frac{3}{16}$ inch thick at the bottom of thread. Some makers prefer to fit thicker tubes, and space them farther apart. For 100 pounds working pressure and upwards, the stay tubes being $\frac{1}{4}$ to $\frac{3}{8}$ inch thick, each alternate tube should be a stay tube; that is, in a nest of, say, 64 tubes, there will be 16 stay tubes.

Stay tubes generally outlast two sets of the ordinary tubes.

Serve tubes, manufactured by John Brown & Co., Sheffield, have a series (6 to 8) of longitudinal ribs running the whole length of the tube and projecting inwards about $\frac{1}{4}$ inch, thus forming an additional absorbing surface for the heat. Besides the additional surface, this tube causes a better circulation of the hot gases within it. In small boilers especially is this tube a useful one, as, with a large grate area, a sufficient heating surface can be obtained.

Retarders.—By introducing into the inside of a plain tube a twisted strip of metal of the same breadth as the inside diameter, Howden found a considerable gain in efficiency, owing to the hot gases getting stirred and circulated in their passage through it.

Stays.—Flat surfaces have to be stiffened and tied together by bars called *stays*. When the plate surfaces are close together and comparatively thin, so that the stays are short and numerous, they are screwed into both plates, and the ends either riveted over or fitted with lock nuts; such stays are usually called "screwed stays." As has been said, the thickness of plates and pitch of stays is interdependent. The size and number of the stays depend on the pressure they have to withstand. The stays in the steam space must be so spaced that a man can pass between them, and for this purpose they should never be nearer than 14 inches, centre to centre, and are usually 15 to 17 inches centres, which gives a clear space of 12 to 14 inches between them. These stays are seldom more than 3 inches effective diameter, and as the exact spacing of them depends on the form and size of the boiler, they are generally arranged to suit the particular case, and the diameter varied to give a section adequate to the load each has to bear. To admit of easy access, these stays are arranged in horizontal and vertical rows as nearly as possible.

The Admiralty allow a stress of 18,000 lbs. per square inch of effective area of stay when at the test pressure, which is usually 90 lbs. above the working pressure. If the stays are below 1½ inches diameter, only 16,000 lbs.

The Board of Trade allow 9000 lbs. per square inch at working pressure.

Lloyd's Registry allow, on stays not exceeding 1½ inches smallest diameter at working pressure, 8000 lbs. on "screwed" stays and 9000 lbs. on others; on stays above 1½ inches smallest diameter, 9000 lbs. on "screwed" stays and 10,000 lbs. on others.

British Corporation Rule is $D = \sqrt{\frac{S \times W}{C}} + \frac{1}{8}$.

S = the surface in square inches supported by the stay.
D = the effective diameter of stay.
W = the working pressure.
C = 7700 for steel screwed stays; 8900 for steel longitudinal stays.

Bureau Veritas has the same rule, but C = 300 × tensile strength of metal in tons, which, for iron, is 22 tons, and for steel according to test.

When of iron, the large stays are often made with a *plus* thread; this necessitates making the ends separate, and welding them to the body, which is of somewhat smaller diameter than at the bottom of thread of the ends. This is a somewhat expensive process, and, as it involves two welds, is not so reliable as simply screwing a rolled bar with a *minus* thread at each end. The latter plan, especially since the making of steel boilers has become general, is now fast taking the place of the former; it has, too, the advantage of excess of section in the body where most corrosion takes place. These stays are secured to the plate with a nut and washer outside, and a nut inside to lock, whose length is two-thirds that of the outside, which is one diameter long. When weight is of consequence, steel bars may be swelled and have a *plus* thread by "upsetting" the ends, as the Board of Trade now permit this.

The screwed stays are usually from 1⅛ to 1⅝ inch diameter, with a Whitworth standard fine thread. The most useful sizes are 1¼, 1⅜, and 1½ inch, suitable for $\frac{7}{16}$ to ⅝ inch plates, and pressures from 60 to 200 lbs. per square inch. When screwed through a plate whose thickness is less than half the diameter, there should always be a lock nut with a thin washer, the nut being two-thirds the diameter of the stay in length. The practice of "nobbling," or riveting over the ends of these stays, is very objectionable when they pass through thin plates, as extreme pressure is very apt to cause the stay to draw completely through the plate, and this is especially so when the plate is ductile and soft like mild steel.

When stays are fitted with nuts and washers at their ends, the following rule holds good :—

$$\text{Pitch of stays in inches} = 10 \times \sqrt{\frac{(\text{Thickness of plates in sixteenths})^2}{\text{Working pressure}}}.$$

Example.—What pitch of stays is suitable for a plate $\frac{5}{8}$ inch thick for a working pressure of 60 lbs.?

$$\text{Pitch} = 10 \times \sqrt{\frac{10^2}{60}}, \text{ or } 12{\cdot}9 \text{ inches.}$$

Board of Trade Rule :—Working pressure $= \dfrac{C \times (T + 1)^2}{S - 6}$.

T being thickness of plate in sixteenths of an inch,

S the surface in square inches,

C being 100 for screw stays with nuts, 165 for longitudinal with nuts and washers two-thirds the thickness of plate.

Lloyd's Rule :—Working pressure $= \dfrac{C \times T^2}{P^2}$.

P^2 being the mean of squares of pitches in rows and between rows, and C and T as before.

C, for screwed stays with nuts, 110 with plates under $\frac{7}{16}$ inch thick; over $\frac{7}{16}$ and under $\frac{9}{16}$ inch, 120; over $\frac{9}{16}$ inch, 135; longitudinal stays, with double nuts and outside riveted washers, two-fifths the pitch in diameter, and half the thickness of plate, 200.

British Corporation Rule :—Working pressure $= \dfrac{C \times (T - 1)^2}{P^2 + p^2}$.

Here P is the greatest and p the least pitch.

C is 330 for screwed stays with nuts, and 476 for stays with double nuts and outside riveted washers, the latter being half the thickness of plate and two-fifths the pitch in diameter.

Bureau Veritas Rule is similar to that of the British Corporation, but the value of C changes with the tensile strength of the plate, and for screwed stays with nuts 415·3 for 27-ton steel, and 461·5 for 30-ton. For longitudinal stays with inside and outside nuts and washers, the outside washers being 0·4 of the pitch in diameter, and two-thirds the thickness of plate, C is 435·4 and 483·8.

Flat plates may be stiffened to allow of wider spacing of the stays than given by this rule, by fitting thick washers of large diameter to each stay, or by connecting the stays to the plates by means of angle-irons or T-bars; the latter plan possesses the advantage of distributing the strain over a large area, and that without a doubtful joint, as is the case with nuts. The old plan of riveting a doubling plate of common iron in wake of the large stays has almost disappeared.

Water Spaces.—The spaces between the furnaces themselves, between the furnaces and shell, and between the combustion chambers, although sometimes diminished, should not be less than 5 inches; that between the backs of combustion chambers and shell should taper from 6 inches at the bottom to 9 and even 12 inches at the top, to allow of the free current upward of the steam generated on the surfaces. If the spaces are less than 6 inches, it is very difficult to hold up rivets, to clean the surfaces from scale, or to get a good circulation.

The space between the nests of tubes should not be less than 10 inches, and, when possible, should be 12 inches. This permits a man to go down to clean across, and ensures good circulation.

Man-holes.—The chief one in the shell should be oval, 16 inches by 12 inches; those in the ends, &c., may be 15 inches by 11 inches; and the smallest through which a boy can pass is 14 inches by 10 inches. Mud-holes are generally 9 inches by 6 inches, and peep-holes 6 inches by 4 inches. The hole cut in the shell plating should be surrounded by a doubling plate or angle bar ring, to compensate for the metal cut away, and the holes through thin plates should have such rings to stiffen the edges. A better plan is to flange the plate inwards so as to stiffen the edge and give a good face for the door joints; this is now very general. The doors are usually placed inside the boilers, and held to their faces by studs screwed into them, which pass through strong cross bars or "dogs," held by square nuts; the main door in the shell is, however, sometimes fitted externally, and connected to a flanged ring with bolts, in the same way as a steam chest door on the engines. The doubling ring is, in this latter case, formed of a very thick angle-iron, whose *deep web* is flanged to fit to the boiler, and whose flange forms the face for the door.

CHAPTER XXI.

BOILER MOUNTINGS AND FITTINGS.

Smoke Box.—This appendage to the boiler is for the purpose of receiving the products of combustion as they emerge from the tubes, and conducting them through the "uptakes" into the funnel. In the old box form of boiler, it was built inside, and formed an integral part of the boiler; but, with the modern cylindrical boiler, it is a separate structure, secured to the boiler front by studs. It is constructed of iron or steel plates and angles of "ship" quality, and made "smoke tight" only; it should, however, be caulked if necessary, to prevent air passing to the inside. In front of the tubes are a number of doors hung on hinges, and so arranged that every tube may be swept or removed in case of necessity. The doors should be of such a size as to be easily handled, and when the nests of tubes are so large as to cause the door to be too ponderous if made in one, two doors may be fitted to close on a portable stanchion. The doors are sometimes arranged to open on a horizontal and sometimes on a vertical axis; the latter is preferable when possible, as then they are more easily handled.

The bottom of the smoke-box should be at least 12 inches broad, measured in direction of the length of the boiler, and, when possible, as much as 15 inches. If too narrow, it is soon filled with soot and ashes, so as to cover the ends of, and render useless, the bottom rows of tubes; the baffle plates on the doors are also soon burned away. The bottom plate should be at least 2 inches below the bottom row of tubes, and the side plate the same distance from the side tubes, so that the tubes may be drawn clear of the 2-inch angle-iron rim around the doorways. The front of the smoke-box is sloped outwards, so as to be about twice the breadth of the bottom from the boiler front, above the level of the top row of tubes. Above this, the smoke-box contracts towards the funnel base, and its configuration must depend on the position of this, and on the consideration that the section transverse to the flow of gases must have an area at least equal to the area through the tubes. The part between the smoke-box and funnel is called the "uptake," or "take-up"; it should have easy bends, and lead as directly as possible to the funnel, and be without recesses and obstacles where the draught may be baffled.

The bottom and sides of the smoke-box and sides of the uptake should be of $\frac{1}{4}$-inch plates for large boilers, $\frac{3}{16}$ for smaller ones, and where weight is a consideration $\frac{1}{8}$ is sufficient. The smoke-box doors should be of the same thickness, and have baffle plates $\frac{1}{8}$ inch thick on the inside, and air or screen plates of the same thickness on the outside; these screen plates prevent radiation of the heat to the stoke-holes, and, for the same purpose, the sides of the smoke-box and uptake should be fitted in the same way. When steel is used, there may be a reduction of 20 per cent. in these thicknesses, and, in naval ships, the outer casings may be $\frac{1}{16}$ inch only. Steel sheets are now cheaper than iron, and will stand flanging, bending, and working in a way that is impossible with iron; consequently iron is now seldom used for such purposes as these.

To protect the boiler front, which has only steam on its inner surface, the uptake should have a back commencing from just above the level of the top row of tubes. When this cannot be done, a good and well-fitting baffle plate should be fixed to the boiler front.

A casing is also fitted round the funnel from its base to the level of the deck casing, to prevent radiation. A corresponding casing is fitted round the funnel, above the level of the deck coamings, to a convenient height, and over this is fitted a hood secured to the funnel, so as to prevent water passing down, while allowing the hot air to come out. This hood is called by various names, as "cravat," "bonnet," &c. (fig. 180). It is now often at the top of the funnel, so the latter is completely protected from the weather (fig. 179).

Where there are several boilers discharging smoke to one funnel, each smoke-box should have a separate uptake, so that the smoke from one does not enter the box of another; and, when there are no good ashpit doors, there should be a damper in each of these uptakes, so as to regulate the draught and get a uniform evaporation from all the boilers, and, in case of necessity, to isolate a particular boiler.

Funnel.—This is usually of circular section, but sometimes, to minimise the transverse size of the boiler-hatch, it is made of oval section. The funnels of men-of-war are often made of oval section for the same reason, but, instead of the section being an ellipse, as is generally the case in the mercantile marine, it is like that of an oval boiler. The best height to look well is 5 to 7 diameters above the taffrail, the latter when there are high bridges or boats in wake of the funnel. For the same reason, the ring for the shrouds should be $\frac{9}{10}$ the diameter from the top. Funnels are made of ship quality plate, lap-jointed, or butt-jointed with single straps inside; the latter costs more, but when so made is more durable. Another method much in fashion at one-time, and which presents a good appearance, is to make the longitudinal joints with inside butt-straps, and the circumferential with a band of iron of a flattened **U** section (fig. 180).

The funnel plates should, for strength, be thicker at the base than at the top, but the top plates wear out faster than those at the bottom. The following may be taken as the approximate thickness of funnel plates:—

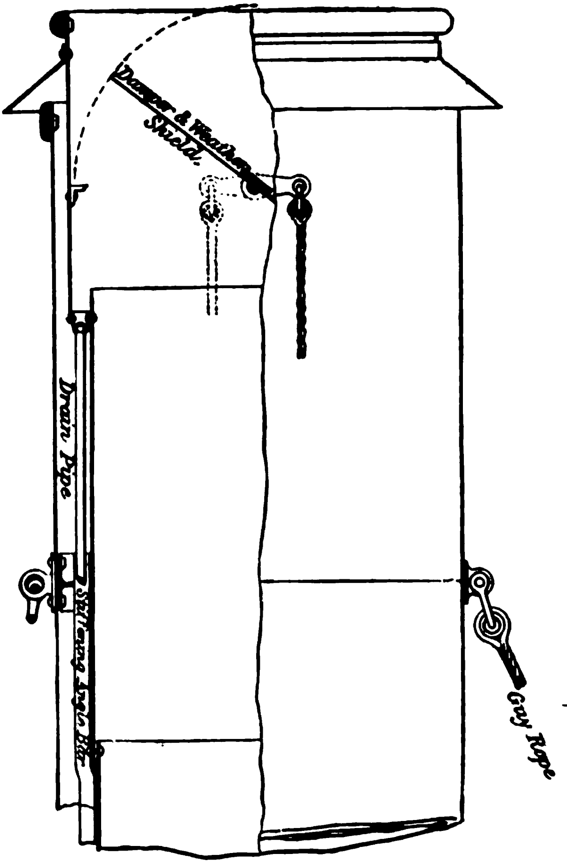

Fig. 179.—Naval and Yacht Funnel Top.

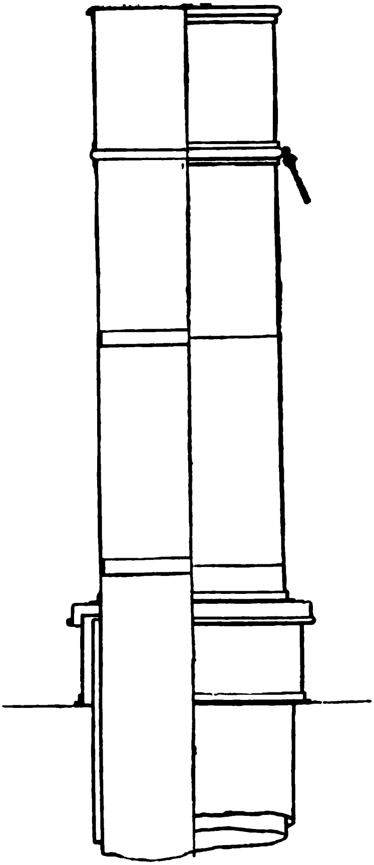

Fig. 180.—Ordinary Funnel, with Hood, &c.

Top plates = 0·1 inch + 0·025 for each foot of diameter.
Middle ,, = 0·125 ,, + 0·026 ,, ,,
Bottom ,, = 0·15 ,, + 0·027 ,, ,,

If the funnel is stiffened with angle or **T** bars (fig. 179), it may be made of somewhat thinner plates. The funnels of naval ships are, of course, made as light as possible, and the plates composing them are seldom more than $\frac{3}{16}$ inch thick, and of steel.

TABLE XLVII.—PITCH, &c., OF RIVETING FOR FUNNELS, CASINGS, &c. (ADMIRALTY WORK).

Description.	Thickness of Plate.	Diameter of Rivet.	Pitch of Rivet.
	Inches.	Inches.	Inches.
Uptakes,	$\frac{3}{16}$ $\frac{1}{4}$ $\frac{5}{16}$ $\frac{3}{8}$	$\frac{3}{8}$ $\frac{1}{2}$ $\frac{5}{8}$ $\frac{5}{8}$	2 2 $2\frac{1}{8}$ $2\frac{1}{8}$
Funnels,	$\frac{1}{4}$ $\frac{3}{16}$ $\frac{1}{8}$ $\frac{1}{16}$	$\frac{1}{2}$ $\frac{3}{8}$ $\frac{3}{8}$ $\frac{1}{4}$	3 2 2 2
Funnel casing,	$\frac{3}{16}$ $\frac{1}{8}$ $\frac{1}{16}$	$\frac{3}{8}$ $\frac{3}{8}$ $\frac{1}{4}$	3 2 2
Deck casings,	$\frac{3}{16}$ $\frac{1}{8}$ $\frac{1}{16}$	$\frac{3}{8}$ $\frac{3}{8}$ $\frac{1}{4}$	2 2 2
Cowls and trunks (exposed to weather),	$\frac{1}{8}$ $\frac{1}{16}$	$\frac{3}{8}$ $\frac{1}{4}$	2 2
Screens and ventilating trunks,	$\frac{1}{8}$ $\frac{1}{16}$	$\frac{3}{8}$ $\frac{1}{4}$	4 3
Other work,	$\frac{3}{16}$ $\frac{1}{8}$ $\frac{1}{16}$	$\frac{3}{8}$ $\frac{3}{8}$ $\frac{1}{4}$	2 to 3 2 to 3 2

Furnace Fronts and Doors.—Although often made of cast iron, they are better made of wrought iron to withstand the rough use to which they are exposed. It is from this cause that all the improved doors which have been tried have been finally rejected; and because of this rough usage all attempts at refinement in the fittings, &c., meet with want of success on board ship. The smaller the door the better, as, when open, an excess of air passes into the furnace and lowers its efficiency; on the other hand, it must be large enough to stoke, work, and clean the fires properly. A long grate requires a larger door than does a short one. Furnaces of large diameter—that is, above 42 inches—should have a pair of doors to be used alternately. The amount of opening when stoking or cleaning fires is thereby reduced, and the sides of the grate are better attended to.

The doors should be so arranged as to remain open in a seaway when required; this may be effected by making a projection and corresponding recess in the hinge. A star damper should be fitted to the fire door, so that, when open, a supply of air is admitted to the fires. The baffle plate inside the door should have a number of small perforations in this case, to finely divide and to distribute the extra supply of air.

Fire-bars.—The length of grate should never exceed twice the diameter of the furnace, as the fires cannot be properly worked when this is the case. To get the highest efficiency of grate, it should not be more than 1½ times the diameter. The slope of the grate should be 1 inch to the foot, which may be increased, in furnaces over 42 inches diameter, to 1½ inch with advantage. If the grate is over 5 feet long, there is generally some difficulty in properly stoking the back end, and it is only a good fireman who can properly work the fire on a long grate. The increased slope materially helps to overcome this difficulty, and at the same time the fire is better supplied with air at the back, and not choked by the products of combustion from the front.

The bridge or brick barrier at the end of the grate should be built to such a height that the area of passage over it is not less than ⅛ nor more than ⅙ the grate area; and, when possible, the distance from the top of the bridge to the top of the furnace should be sufficient for a man to pass into the combustion chamber.

When South Wales or other similar coal is to be burnt on the grate, there should be no space between the bars and the side of the furnace; when the furnaces are corrugated, these side bars should be made to fit into the corrugations.

The fire-bars are usually in two lengths; but the grate is more efficient when they are in one, as the bearer is avoided, which baffles the free flow of air to the fire above it, and prevents the fireman from "pricking" effectively. Cast-iron fire-bars to burn bituminous coal may be 5 feet 6 inches long, but if the coal contains much sulphur they are safer in two lengths. The Admiralty do not allow the bars to be longer than 27 inches, even when made of wrought iron, as they generally burn Welsh coal, and may have to use, on foreign stations, coal containing sulphur.

Fire-bars are made from ⅞ to 1¼ inch broad on the face; the former is better when the bars are not very long, and when of wrought iron may with advantage be even ⅝ inch. In the mercantile marine 1¼ inch is the usual breadth when the bars are long.

The depth of the bar at the middle depends on the length, and should be:—

$$= 0{\cdot}6\sqrt{\text{length}},\ \text{when of cast iron,}$$

and

$$= 0{\cdot}5\sqrt{\text{length}},\ \text{when of wrought iron.}$$

The thickness at the bottom should be one-third the breadth at the face, and should taper to two-thirds beneath the flange.

For burning bituminous coal there should be a space of ½ inch at least between the bars, and, when it cakes quickly, there may be as much as ¾ inch; but, if Welsh coal or American anthracite is to be burnt, there should never be more than ½ inch spaces, and with narrow bars the space may with advantage be less.

Martin's Patent Bars consist of wrought-iron bars of square section placed with the angles upward, and so arranged that the bars may be slightly turned so as to clean the fires.

Henderson's Patent Door and Bars.—This is one of the most successful of the improved grates. The bars are of cast-iron and ordinary section, hung in a frame, which can be easily moved by means of levers, so as to slightly move alternate bars longitudinally; this movement, while carrying the fuel gradually to the back of the furnace, breaks up the clinker, and obviates the necessity of cleaning the fire. The door of the furnace is hinged horizontally at the bottom, and so arranged as to drop into a recess in the dead plate left for it, which recess is filled by a back piece attached to the door, and turns down as the door opens; if the fire needs cleaning, the door is turned still farther, so as to leave the gap in the dead plate, and form a slope below it to shoot the clinker and cinders into the ash-pit instead of raking them on to the stokehole floor. The furnace front, too, is carefully designed, so as to pass a current of air completely about it between the front and back, thus serving the double purpose of keeping the front comparatively cool, and of heating the air before entering over the fire. This arrangement of grate permits of a rapid and complete combustion of coal equivalent to a moderate forced draught, as may be seen by referring to Tables 1*d* and 1*e*, where grates are marked "H."

Liquid Fuel.—To burn petroleum refuse and other similar liquids, a special burner is necessary, and on its efficiency depends the economy of using such fuels.

The functions of such an apparatus are to direct the fluid in a steady stream to the place where ignition commences; to pulverise it, or "spray" the stream so as to permit of quick and perfect ignition; to project the flame in the required direction, and to keep it constantly going with the least possible expenditure of steam, or anything else which costs money.

Mr. James Holden, the well-known locomotive-superintendent engineer of the Great Eastern Railway, after experimenting for a considerable time with the the burners used on the Caspian Sea and in America, devised an instrument of his own superior to any of them, which has been found to give the greatest satisfaction, both on locomotives and on shipboard, by performing its duties efficiently and without fail over considerable periods. Steam is employed to drive the liquid through the nozzle (fig. 129), but, before doing so, it has induced a flow of air in with it from the central tube, which air mingles with the pulverised liquid and helps to burn it completely, as well as to furnish the oxygen required for ignition. The ring around the nozzle is charged with steam, and this forms a heater for the air current, which flows past it to the nozzle. There are, however, other burners which, if not so efficient as this, have the merit of not requiring steam to work them (*vide* Sir F. Flannery's paper in *Trans. I. Naval Architects*, vol. xliv.).

Stop-valve.—The main stop-valve should be of sufficient size to pass out all the steam the boiler is capable of making *with little resistance*, but no more than all. The chief loss of pressure in the cylinders is often due to the stop-valves being only partially open; on the other hand, when the cylinders are large and the strokes of the piston comparatively slow, priming may be effectively checked by partially closing the stop-valve, so that there is no sudden withdrawal of steam, *in fact, the valve should not pass out more steam than the boiler is making.*

The area of opening, sufficient to pass all the steam a boiler can make can be obtained from the following rule:—

$$\text{Area of orifice} = \frac{\text{total heating surface} \times \text{R}}{8 \times \text{P}}.$$

R is the rate of evaporation per square foot of heating surface, P is the pressure absolute.

The area through a pipe to carry away the steam is two to three times the net area, depending on the length (Table xiv*a*).

The diameter of the stop-valve is generally settled from considerations of the size of the main steam pipe at the engines. Let D be the diameter of the main steam pipe, and n the number of boilers (at least two), then

$$\text{Diameter of branch pipe to each boiler} = \text{D}\sqrt{\frac{4}{3n}}.$$

The area of pipe section to suit a boiler may be found by the following rule:—

$$\left.\begin{matrix}\text{(0·25 square inch per square foot of grate + 0·01 square} \\ \text{inch per square foot of total heating surface)}\end{matrix}\right\} \times \sqrt{\frac{100}{\text{pressure}}}.$$

Example.—To find the diameter of steam pipe from a boiler whose grate area is 50 square feet, and the total heating surface is 1500 square feet; pressure, 80 lbs.

$$\begin{aligned}\text{Area of section} &= (0{\cdot}25 \times 50 + 0{\cdot}01 \times 1500) \times \sqrt{\frac{100}{80}} \\ &= 30{\cdot}7 \text{ square inches.}\end{aligned}$$

Therefore the diameter should be 6¼ inches.

Example.—To find the diameter of the main steam pipe of a locomotive boiler whose grate area is 16 square feet, the total heating surface 1200 square feet, and the pressure 150 lbs.

$$\begin{aligned}\text{Area of section} &= (0{\cdot}25 \times 16 + 0{\cdot}01 \times 1200) \times \sqrt{\frac{100}{150}} \\ &= 13{\cdot}09 \text{ square inches.}\end{aligned}$$

Therefore the diameter should be 4⅛ inches.

The diameter of the stop-valve should be such that the clear area past it is not less than given by the above rules.

$$\text{The diameter of spindle} = \frac{\text{diameter of valve}}{50} \times \sqrt{\text{pressure}} + \tfrac{1}{8}\text{ inch.}$$

The valve and seat are of a bronze which should be both hard and strong at the temperature of the steam passing it (Chapter xxv.). The valve should have the boiler pressure always on the side opposite the spindle, so that it helps to open it. The spindle should have a square thread, and, when possible, the screwed part should be outside, so that in opening or shutting the valve the spindle does not turn round. As the full pressure is on the valve when shut, the bridge, &c., should be strong enough to withstand it. The seat, when fitted with wings for the guide to the spindle, should be carefully secured and the wings curved, so that, when expanding with the heat, the seat is not distorted. These seats, when fitted into cast iron, are very apt to get loose and leak from the permanent set of the metal, induced by the resistance of the cast iron to expansion of the bronze. They should, therefore, be secured by bolts or studs to the iron casting, and a joint made which will keep tight in spite of expansion, &c., much in the same way as the pipe joints are done.

The Admiralty require stop-valves and all other boiler mountings to be made of bronze. This, no doubt, is a wise precaution, though it has not always prevented, but in some cases has been the cause of, accident.

The Admiralty likewise require all stop-valves to be self-acting, that is, they shall close on the pressure in the boiler being decreased below that in the main steam pipe. This is with the object of localising the danger and disconnecting the boiler in case of accident to it from shot, &c. Such valves have their spindles turned down at their outer ends, so as to pass through a hole in the screwed shank, fitted to the cross-piece, as shown in fig. 181, and with cross-handles to permit of the valve being turned or closed as required.

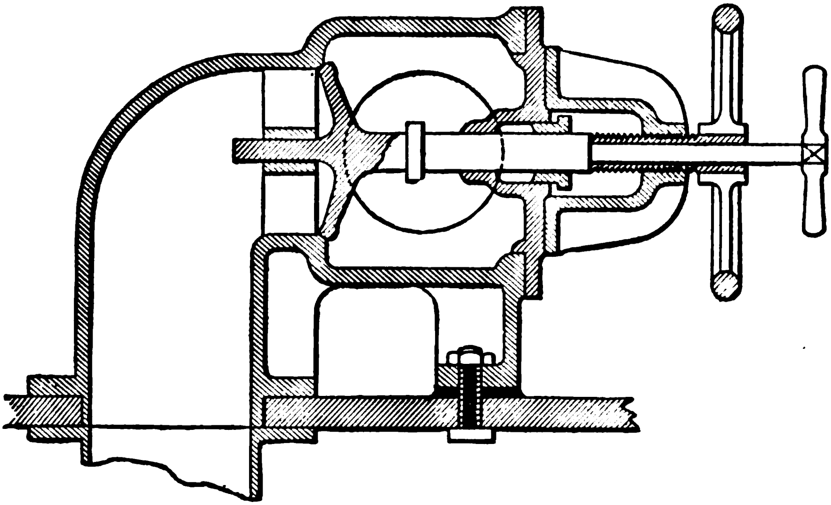

Fig. 181.—Naval Boiler Stop-valve.

Safety Valve.—As its name implies, this valve is for the purpose of providing a safe and self-acting means of relieving the boiler from excessive pressure. A good safety valve should be (1) large enough in diameter, and have sufficient lift to allow the steam to escape as fast as it is generated, when the pressure is 5 per cent. above that to which the valve is loaded; (2) it should be so made that it closes again as soon as the pressure has dropped below the load; (3) it should be free to open and shut, so that it may always act efficiently and promptly; (4) it should be so enclosed that it cannot be tampered with or accidentally interfered with by pieces of coal, &c., falling into it; and (5) for marine purposes it must be so constructed as not to be affected by the motion of the ship.

It is unnecessary to deal with weight-loaded valves, as none are now used, the fifth condition being satisfied by means of steel springs for the load (fig. 183). When weights were used, the amount of lift given to the valve by the steam pressure was very small, and, since the lifting of the valve compresses the spring and increases the load, the spring-loaded valve opens less. This being so, area of opening can only be obtained by increasing the diameter of the valve. Many ingenious methods of increasing the lift have been tried, but all those involving the use of special mechanism have given place to those which do without it; the most successful and best known of the latter is Richardson's Patent (fig. 183), generally called *Adams'*, after the name of the manufacturer, who purchased, improved, and worked the patent in this country, and Cockburn's Valve (fig. 184), designed and made by Messrs. Cockburn, of Glasgow. These consist essentially of an ordinary mushroom valve with a secondary outer rim of U section which overlaps the rim of the seat, so that there is a second

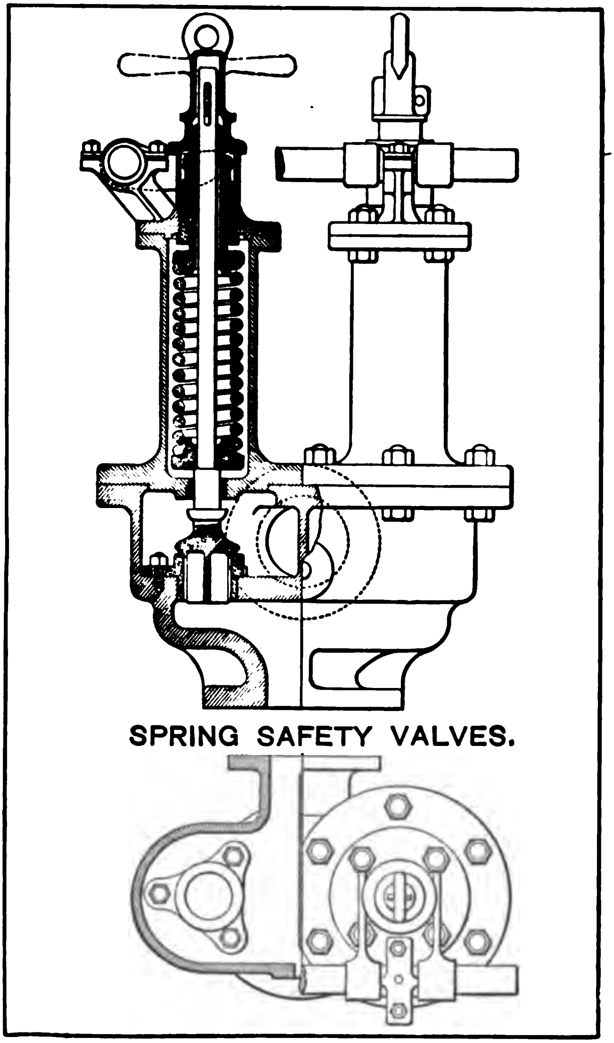

Fig. 182.

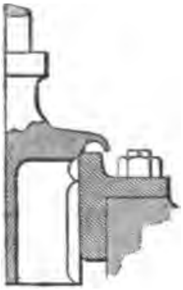

Fig. 183.—Adams' Valve.

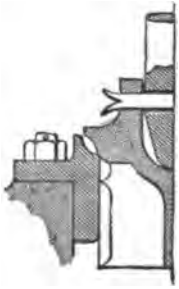

Fig. 184. — Cockburn's Valve (Navy Type).

contracted orifice at the outer edge of this rim. As soon as the valve opens, the steam fills the outer rim, and the valve is then virtually of larger area; the load on it is so suddenly increased that the valve lifts wide open immediately, and will continue to vibrate with the spring until the pressure falls so as to be insufficient to open the valve when the latter touches the seat in one of its vibrations. The second condition is best fulfilled when the valve can be made to dance over its seat from the vibration of the spring. The third condition can only be fulfilled by making every movable part a very easy and, in most cases, *a very slack fit*, and having springs whose resistance to compression is uniform on every side; this is a quality not always found in the springs supplied commercially, and adjusting a spring when made is expensive and troublesome; to avoid the evils due to this cause, Messrs. Cockburn have devised a joint for the spindle which shall adjust itself to suit the irregularity of pressure when the spring is compressed (fig. 184*a*).

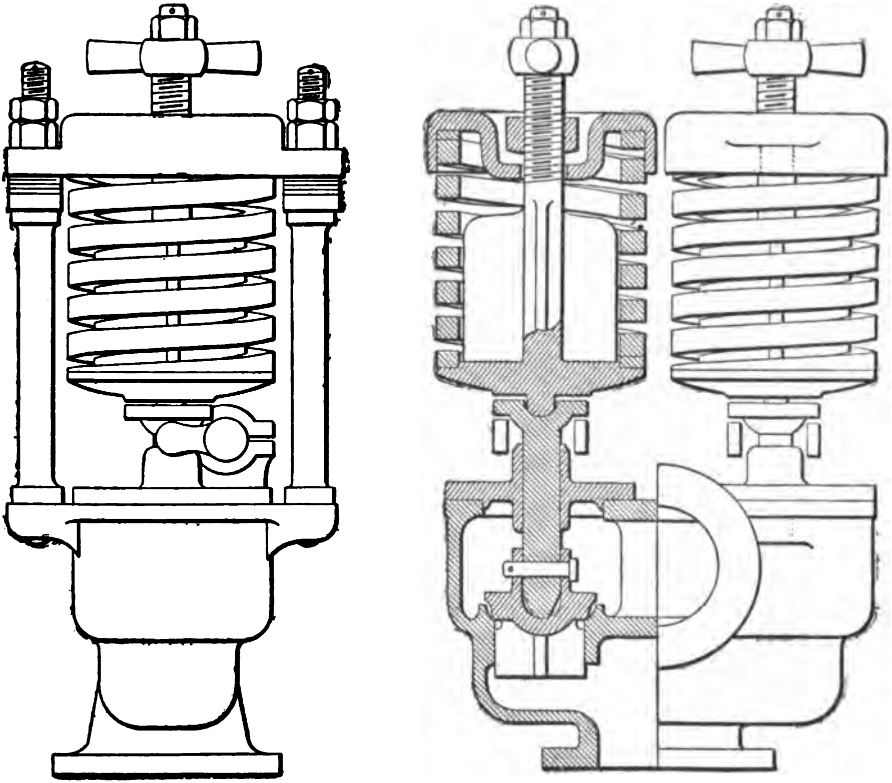

Fig. 184*a*.

To prevent the valve from being injured by accident or design, it should be enclosed in a case, and the Board of Trade require that such cases shall be locked up, and the key kept by the captain of the ship.

The Size of Safety Valve.—This depends on the *volume* of steam which can be generated by the boiler in a given time, and that depends on the weight of fuel it can consume, on its efficiency, and on the working pressure. In similar boilers—that is, boilers made on the same general design and for the same pressure—this volume of steam varies with the grate area. The original rule laid down by the Board of Trade for the area of safety valve is based on this, and it was found to work satisfactorily; since then, however, new rules have been laid down. Strictly speaking, any rule for the safety valve should fix the amount of *circumference* rather than area, and considerable allowance should be made for the load pressure, as for the same weight of steam the volume varies inversely as the pressure.

The following are the rules for the size of valve:—

(1) To satisfy the *Board of Trade*, the valves must not be in any case less than 2 inches in diameter, and for each boiler with natural draught, the area of valve or valves combined must not be less than given by this rule.

$$\text{Area of safety valve in square inches} = \frac{37\cdot5}{\text{P}} \times \text{grate area (square feet)},$$

P being the *absolute* boiler pressure. For forced draught the area found by the above rule must be multiplied by $\frac{\text{C}}{20}$, where C is the estimated coal consumption in lbs. per square foot of grate per hour.

(2) *Lloyd's* retain the old Board of Trade Rule that the valve area must be at the rate of ½ square inch for each square foot of grate, but allow special valves of any size, so long as they be satisfactory when tested.

(3) The *French Government Rule* is based on the amount of heating surface contained in a boiler, and this, perhaps, is the truest gauge of a boiler's capability, as it bears a constant relation to the amount of coal consumed; allowance is also made for the steam pressure. The diameter is in inches, the heating surface in square feet, and the pressure in lbs. per square inch.

$$\text{Diameter of valve (if only one)} = 1\cdot23\sqrt{\frac{\text{total heating surface}}{\text{pressure} + 9}}$$

(4) The *German Government Rule* also makes allowance for the steam pressure, and is as follows:—

To have a *clear* area of valve or valves, after deducting for the wings or other obstacles, at the rate of so many *square millimetres* for each square metre of total heating surface, in accordance with the following table:—

Working pressure in atmospheres	5	6	7	8	9	10
Number of square millimetres per metre of surface	131	112	98	86	79	72

Working pressure in atmospheres	11	12	13	14	15	16
Number of square millimetres per metre of surface	66	60	56	54	52	51

(5) An improved rule, which is simple and easily used, is—area of each of two valves = (0·05 square inch per square foot of grate + 0·005 square inch per square foot of total heating surface) × $\sqrt{\frac{100}{\text{pressure}}}$.

Example.—To find, by the various rules, the size of a pair of safety valves for a boiler whose grate area is 45 square feet, heating surface 1500 square feet, and working pressure 180 lbs.

(1) By Board of Trade.

$$\text{Area} = \frac{45}{2} \times \frac{37·5}{195} = 4·33 \text{ square inches.}$$

Therefore *the diameter is* $2\frac{3}{8}$ inches of each valve.

(2) By French Government rule.

$$\textit{Diameter} = 1·23\sqrt{\frac{1500}{180 + 9}} = 3·47 \text{ inches.}$$

(3) By German Government rule, for one valve.

$$\text{Clear area} = 60 \times 1500 \div 10·76 = 8365 \text{ square millimetres.}$$

$$\text{Net area} = \frac{8365}{645} = 13 \text{ square inches.}$$

Add to this 2 square inches for obstruction of wings,

gross area = 15 square inches,

and diameter for one valve is 4·4 inches, for each of two valves 3·1 inches.

(4) By the improved rule.

$$\text{Area} = (0·05 \times 45 + 0·005 \times 1500) \times \sqrt{\frac{100}{180}} = 7·3 \text{ square inches.}$$

Therefore *diameter is* 3·1 inches.

The mitre on a safety valve-seat should not be more than $\frac{1}{16}$ inch broad, except for very large valves, and the bearing area in any case need not exceed that necessary for a pressure of 1200 lbs. per square inch on it when there is no steam pressure on the valve; hence

$$\text{Breadth of mitre} = \text{diameter of valve} \times \frac{\text{working pressure}}{4800}.$$

The Board of Trade rule for the size of steel for the spring is

$$d = \sqrt{\frac{S \times D}{C}}.$$

S is the total load on the valve; D the diameter of coil, measured from centre to centre of wire, in inches; d is the diameter of round wire, and the side of square section wire; C is 8000 when the coil is made of round section steel, and 11,000 when of square section (*vide* Appendix D).

Internal Pipes should be fitted from the stop-valves to the highest part of the boiler, and be made with holes or slits, whose collective area is equal to twice the area of section of the pipe.

The chief object of this pipe is to collect the steam gently from every part of the boiler, so as to avoid setting up a strong current in one particular direction, and thereby induce priming. These pipes are usually made of brass, but some engineers prefer copper, and others make them of cast iron to avoid risk of galvanic action and reduce the cost. By fitting an internal pipe, the stop-valve can be placed in a position convenient for examination and working and it should always be so situated as to be easy of access at all times. Arrangements should also be made for opening and shutting it without going into a position of danger or difficulty, and this can always be effected by lengthening the spindles or fitting chain gear. The Admiralty and some passenger-ship owners insist on having gear fitted so that the stop valves can be shut from on deck as well as in the stoke-holes.

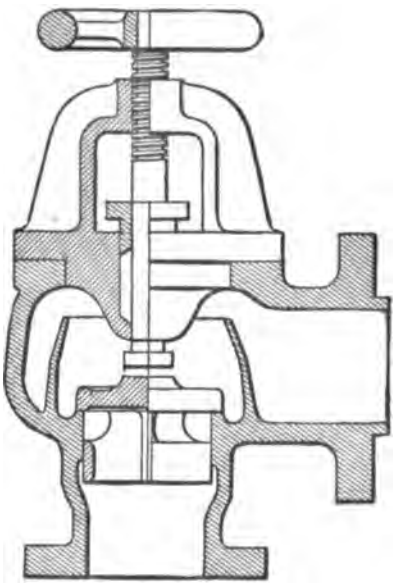
Fig. 185.—Improved Feed Check Valve.

In the mercantile marine, the stop- and safety-valve boxes are almost invariably made of cast iron; the valves, seats, and spindles being of bronze. The Admiralty require all boiler mountings to be made of bronze or cast steel, and do not allow cast iron to be used.

Feed-valves.—Each boiler should be fitted with a self-acting, non-return valve, through which the main feed-water is pumped. It should also have a screw spindle, which may be used to regulate the lift, or to shut it down when water is not required. There should also be a similar valve through which the auxiliary pump can discharge water to the boiler.

The valve is generally of mushroom form, and made similar to the ordinary stop-valve, except that it is detached from the spindle. It is made wholly of gun-metal, and should be very strong, as at times the pressure on it may be excessive. An inner casing should surround the valve (fig. 185), and be formed or pierced in such a way that the flow is fairly even round the periphery, otherwise the valve tends to cant and wear its seat unevenly, with leakage as a consequence. Locomotive engineers sink the feed check valve seat well below the discharge orifice for this purpose.

There should be 4 square inches of clear area through the valve and pipe for every 100 lbs. of water evaporated per minute; or, put in a more convenient form—

Area through main feed-valve in square inches
= total heating surface in square feet ÷ 250;

and, area through donkey feed-valve in square inches
= total heating surface in square feet ÷ 300.

As the feed-valves cannot always be placed on that part of the boiler best suited to receive the feed-water, and also in order to distribute that water so as to avoid its affecting the boiler plates, an internal pipe should be always fitted. To avoid the necessity of blowing the boiler down in case of accident to the feed-valves, it is a very common practice to fit these valves high up on the boiler, even in many cases above the water-level. This plan also has the advantage of providing a means of warming the feed-water, than which nothing is more essential for the preservation of the boiler; the heating is effected by the passage of the water through a long internal pipe of brass or copper, which leads it to where there is a down current of water, so that the comparatively cold feed-water may not interfere with the circulation. To permit of the check valves being examined and ground in without emptying the boilers, a stop valve is fitted, as shown in fig. 185*a*, and proved to be very useful, especially in ships making long voyages.

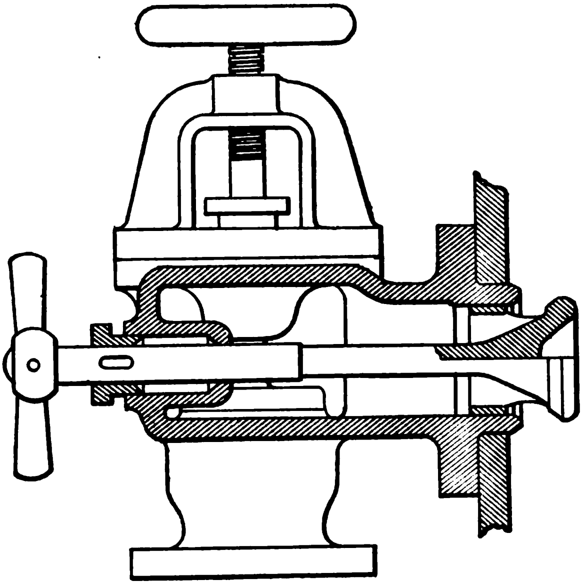

Fig. 185*a*.—Feed Stop and Check Valve, Combined.

Some engineers prefer to inject the feed-water in the form of spray, either above or a little way beneath the surface of the water in the boiler; this avoids all chance of injury to the boiler plates, as any gaseous matter mechanically mixed with the feed-water is at once given up and mixes with the steam.

Great care should be taken in any case that the internal feed-pipes "run full"; that is, that they are never at any time filled or partly filled with steam, but always with water.

The *dynamic* effect of the steam on the feed-water, when mixed inside the pipe, is very startling; every stroke of the feed-pump delivering cool water produces a concussion by the sudden condensation of the steam, so that, in a very short time, both external and *internal* pipes are damaged seriously

To avoid this, the internal pipe should, when discharging above the water-level, be *turned upward* at the end, so as to always remain filled with water; and when turned downward to discharge under water, the end should be well below the lowest working level.

An additional means of safety is sometimes afforded by fitting inside the boiler a clack valve, so arranged as to close over the end of the internal pipe or on the spigot of the ordinary check valve; when this is provided, the latter can be examined when steam is up. A cock is also sometimes fitted close to the check valve, so that the supply can be regulated by it, instead of by interfering with the lift of the check valve.

Blow-off Cock.—A cock used to be fitted at or near the bottom of the boiler, to answer the double purpose of admitting sea-water before getting up steam and to *blow off* some of the water when required. This cock should be a very strong one, as it is liable to rough usage, and, being out of sight and not easily got at, it is very apt to be neglected. For this reason, as well as because a large cock is difficult to open and shut, some engineers prefer a valve to a cock. If a cock is fitted, it should be so arranged that its handle or spanner cannot be removed when it is open. Sea-water is now seldom or never used for filling boilers working at high pressures, so that a connection with the sea is not necessary, but when it is the clear area through a blow-off cock should be = 1 square inch + 0·2 square inch for each ton of water in the boiler.

As it is a very reprehensible practice quickly to blow off a marine boiler when at its normal working temperature, a somewhat smaller cock is fitted to the bottom of the boiler, so that, when the pressure of the steam is down, the water may run into the bilges and be pumped overboard.

Scum Cock.—A cock, having a clear area through it of one-third that of the blow-off cock, should be fitted to the boiler, near to the level of the water, and to it is connected a perforated pipe, inside the boiler, not lower than the lowest working level. The object of this pipe is to collect all *scum* and floating impurities from the water and discharge it overboard. Considerable quantities of grease and greasy matter are pumped with the feed-water into the boiler, and should be got rid of occasionally. A particular kind of hard grease is sometimes formed in the condensers of engines whose cylinders are lubricated with a certain class of oils which are not pure hydrocarbons; portions of it are pumped with the feed-water into the boiler in the form of small pellets, which, being of superior specific gravity to pure water, sink to the bottom, and remain there until the density of water increases sufficiently to cause it to rise and come in contact with the hot surface. It is, therefore, better to filter the feed-water under these circumstances than to trust to a scum cock or even to a blow-off cock, and to use little or no internal lubricant; and, if a lubricant is used at all, to be careful it is specially suited for the purpose and a pure hydrocarbon (*vide* Chapter xxvi.).

The scum cock was used as a means of reducing the quantity of water in the boiler before adding a fresh supply from the sea; but, if the surface was clear of dirt, this was better done with the bottom blow-off, especially if it was possible to check evaporation for a few minutes before blowing off.

Water Gauge.—It is of the first importance that those in charge of a boiler shall know with certainty the position of the water-level within the boiler. The ordinary gauge for this purpose consists essentially of a glass tube, whose ends communicate freely with the inside of the boiler, and so situated on the boiler that the plane of the water surface bisects the tube transversely when at its normal working level. It is, however, found necessary in practice to use considerable discretion in the choice of position of this gauge. Since a difference of one-tenth of a pound pressure corresponds to 2·7 inches of water, it is quite possible so to place the gauge as to give very false readings. The upper end of the gauge should not communicate with the boiler near to any exit for steam, for the rush of steam past the orifice can easily make a reduction in pressure of one-tenth of a pound in the gauge pipe. The lower end should also be clear of any part from which steam is evolved, as steam bubbles might flow into the pipe, and tend to raise the water-level in the glass. It should also be well away from the currents due to circulation or other causes, and as near to "dead" water as possible.

It is usual, especially with large boilers, to fit the gauge cocks and *test cocks* to a brass casting connected by pipes to the bottom and top of the boiler; this is called a "stand pipe," and is a necessity when the gauge is on the front of the ordinary boiler. The test cocks are placed on the stand pipe at the lowest and highest working levels, for the purpose of checking the glass gauge, and for use when the latter is broken or out of order. In these days of high pressures, however, they are practically of no use, and might well be done away with and their price go towards an additional glass gauge.

The length of the gauge glass visible should be at the rate of 1¼ inch for each foot of diameter of the boiler; the external diameter of the tube is ⅝ inch for small boilers and ⅞ inch for large ones; the glass is usually about ⅛ inch thick. The Admiralty use ⅝-inch glasses for all sizes of boiler.

The gauge is so placed that the water is just disappearing, or, as it is generally said to be just "in sight," when the level is from 2 to 4 inches above the top of the combustion chamber; the allowance should be 0·3 inch for each foot of diameter of boiler.

The pipes connecting the stand pipe to the boiler should be from 1 inch to 1½ inch diameter, and of strong copper, so as to be fitted direct to the boiler. The Board of Trade and some engineers insist on having a cock on the boiler at the top and bottom; but this, like many other intended extra safeguards, is itself a real source of danger, for the cocks are apt to be shut by mistake or carelessness, and thus cause the gauge to show a false level. That this is no mere fanciful danger has been proved on more than one occasion.

All large boilers, and those in ships which are often under sail, should have two water gauges placed as far apart as possible in an athwartship vertical plane.

The gauge and test cocks should always be fitted with a small plug in line with the bore, which, on being removed, allows a wire to be introduced to clean it of deposit and scale.

Some engineers have the stand pipes and connecting pipes so arranged that there is a continuous flow from top to bottom *only via the glass tube*, the stand pipe being divided in the middle, or being so formed that the middle is only a connecting bar for top and bottom to prevent the glass tube being drawn out. This is very simple, and safer than the ordinary method.

Steam Gauge.—The steam gauge on Bourdon's principle is now nearly universal and so well known as to need no description. Schœffer's original gauge, although less liable to derangement than Bourdon's, is not so accurate, and does not find so much favour. The boiler gauge should have a dial so marked that it may register pressures to at least 25 per cent. higher than the working pressure of the boiler. These gauges should be carefully tested when new, and at frequent intervals after being at work, as it is often found that they require some slight adjustment.

Sentinel Valve.—The Admiralty used to require each boiler to have a small valve loaded with a weight to a few pounds per square inch above the working pressure, so that in case of the safety valves sticking fast and the gauge being false, an alarm may be given when there is an excess of pressure. Such valves are generally about $\frac{3}{4}$ inch in diameter, but sometimes as small as $\frac{3}{8}$ inch. An arrangement of a small safety valve attached to a whistle has been introduced, but there should be no necessity for such refinements, and it is doubtful if, in time of need, they would be heeded.

Weir's Hydrokineter.—This instrument (fig. 186) is for the purpose of warming the water in the bottom of the boiler when getting up steam. It consists of a series of nozzles, one within the other, each having a grating-body in rear, through which the water passes on its road to the nozzle, when a current is set up by a jet of steam issuing from the centre one. The steam is obtained from the auxiliary boiler, which has been used to supply the winches. Without this instrument the bottom of a large boiler remains cold, even after the steam is raised; with it the temperature of the water at the bottom differs very little from that at the top; steam can in this way be safely raised in a shorter time than usual, and at no extra cost, and the endurance of the boiler is very considerably increased. There are many other ways of promoting the circulation when steam is up, but none do this so efficiently during the time of raising steam as the *hydrokineter*.

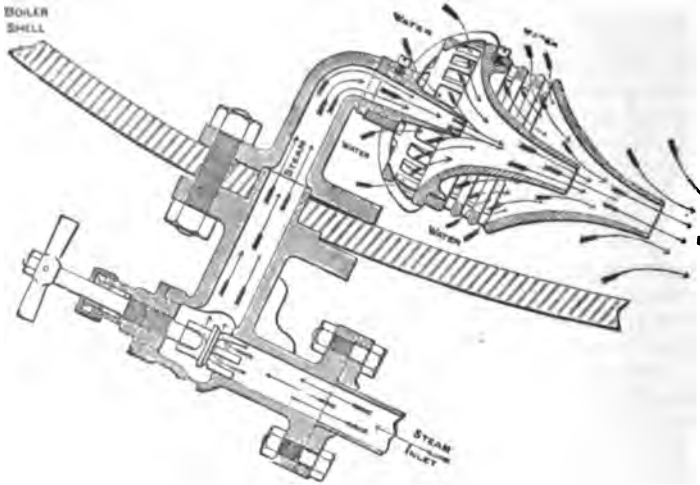

Fig. 186.—Weir's Hydrokineter.

Steam Whistles are of two kinds, known as the bell-whistle and organ-tube whistle; the latter has now superseded the former, on account of its simplicity of construction and superior tone. An improved form has a division in the tube, or two separate tubes, so as to emit two distinct notes which may be in harmony or discord, and when sounded together are heard a long distance.

It is important that the whistle shall sound as soon as the steam is turned on; to insure this happening, great care must be taken to keep the whistle-pipe free of water, which is no very easy matter. It may, however, be effected in two ways: first, by leading the pipe from the boiler into the funnel, and keeping it inside as far as the level of the whistle; second, by taking steam for the steering engine from the top of the whistle-pipe, thereby ensuring a constant flow of steam and no accumulation of water.

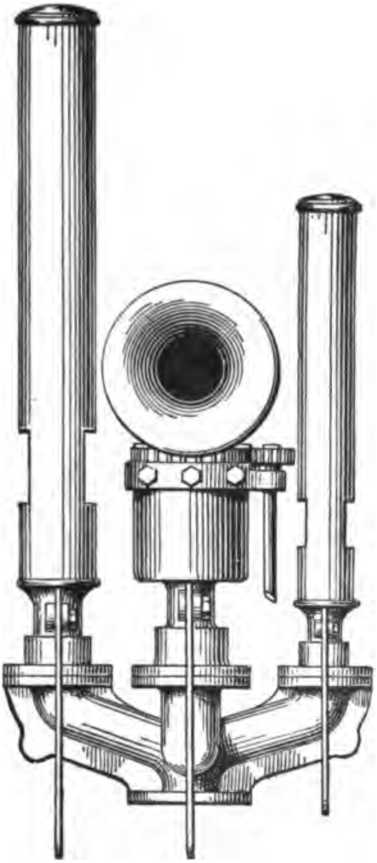

Fig. 187.—Combined Syren and Double Organ Whistle.

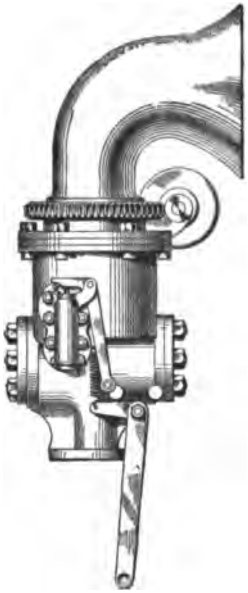

Fig. 187*a*.—Steven & Struthers' Admiralty Syren.

Separator.—This, although not a boiler fitting, is intimately connected with them; it is almost unknown in the mercantile marine, although it might be used sometimes with advantage there in order to free the steam from water mechanically mixed with it. All men-of-war were formerly fitted with separators, and, from the tendency to prime on the part of their boilers when working at full speed, and the danger to the engines when working at a high velocity of piston, from water getting into the cylinders, they were necessary. Now, only ships with certain water-tube boilers have separators.

The old separator consisted of a vertical cylindrical chamber, having a division-plate extending from the top to about half-way down, and so placed that the steam in going through the separator must pass under this diaphragm; the object is to separate out the water mechanically mixed with the steam, by dashing it against the diaphragm, and precipitating it to the bottom of the separator, whence it is blown to the hot-well or sea, whichever is convenient. Most modern separators are constructed on the centrifugal principle, so that the water by its great density follows one course, while the steam follows another.

Boiler Clothing.—The boiler shell should be well covered with a coating of non-conducting material, to prevent loss by radiation from its surface, which may amount in some cases to more than 10 per cent. The material used should be a non-conductor of heat as well as incombustible and inorganic. The following materials* are those in general use for boiler clothing :—

Silicate Cotton, manufactured from slag and having the appearance of cotton, is eminently fitted for boiler clothing. It is a good non-conductor, incombustible, and imperishable from chemical action; it is, however, very brittle, and for this reason will not withstand mechanical action; and, therefore, if loosely packed and subject to vibration, it soon becomes dust, which is most offensive if it gets into the engine bearings.

Asbestos Fibre has very much the same nature as silicate cotton, it is not so efficient, but is more durable. A paste made with this material and magnesia is very efficient.

Magnesia, either alone as cement, or mixed with asbestos fibre into a paste, is a very excellent covering, and, moreover, is very light in weight, so that an extra thickness of it weighs no more than does an ordinary coat of other compositions of a similar nature.

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* Some experiments on the efficiency of different heat insulators are described by Mr. S. H. Davies in a recent issue of the *Journal of the Society of Chemical Industry*. The following table shows the heat transmitted through 1 foot thickness of the different materials for each 1° F. of temperature between the hot and cold surfaces :—

Air-Dried Material.	B.T.U. per Square Foot per Hour.	Weight of Medium per Cubic Foot.	Cost per Cubic Foot.
		lbs.	d.
1. Slag wool (light),	0·054	8·6	8·3
2. Hair felt,	0·058	7·9	20·6
3. Light magnesia,	0·062	10·1	26·0
4. Granulated cork,	0·069	6·1	9·8
5. Slag wool (heavy),	0·070	32·9	31·8
6. Kieselguhr,	0·073	15·0	9·6
7. Flaky charcoal,	0·082	14·5	7·8
8. Pumice (⅛-inch mean diameter),	0·095	25·0	24·1
9. Sawdust (spruce),	0·096	13·1	0·5
10. Asbestos fibre,	0·136	14·5	137·0
11. Sawdust (very moist),	0·298	...	...

Mr. Davies notes that powdered pumice is much inferior to pumice in small lumps.

Cements of various kinds are used, their efficiency depending generally on the amount of vegetable fibre contained in them. These have not so high an efficiency as the foregoing.

Fossil Meal, or infusorial earth, is a composition containing large quantities of minute shells, and forms a most efficient covering; besides being inexpensive, it is also incombustible and durable, and comparatively light.

Papier Maché is employed for this purpose, and is a fairly good material, but it is not altogether incombustible and is apt to rot, besides being heavy.

The last three materials must be put on when the boiler is hot, and be carefully done; this does not militate in their favour, as it is an objectionable thing to have steam on the boilers when the ship is being finished.

The silicate cotton and asbestos fibre may be covered with sheet iron. The cements are generally tarred over so as to be waterproof, and the parts exposed to wear covered with sheet iron or lead.

No wood should be used for clothing when it is possible to avoid it, as it so soon rots, and is at all times liable to take fire.

Cameron's Patent Lagging is an ingenious arrangement of wrought-iron framework, *strapped* to the boiler, so as to form a series of segments, which may be filled with any non-conducting substance, and is covered in with squares of corrugated sheet iron, secured in such a way as to be easily and quickly removed for examination or repair. This is the most perfect plan, as it avoids all piercing of the boiler shell with studs and the use of combustible materials, and while capable of being well secured, it is such as to be wholly removed and replaced in a very short time.

A much thicker coating of lagging is necessary with the high temperatures now common, and no boiler, pipe, or cylinder can be considered properly clothed if the temperature at the surface is more than 30° F. above that of the atmosphere surrounding it. The bottoms of boilers should be as well covered as the tops; in fact, no heat should be allowed to escape to the air from any part of the boiler that can be kept in. Even the uptakes and smoke boxes may, with advantage, be coated.

CHAPTER XXII.

FITTING IN OF MACHINERY, STARTING AND REVERSING OF ENGINES, ETC.

Fitting Machinery into the Ship.—As soon as the building of the ship is sufficiently advanced to allow the engineers to commence their work, a line should be stretched in the place intended for the axis of the shafting, and from it reference lines must be scored on the bulkheads, sternposts, and other convenient places for future guidance, and to enable the shipbuilders to set the engine seating and tunnel pedestals with some degree of accuracy.

For this purpose piano wire answers best, as it can be drawn exceedingly tight without breaking, and the amount of "sag" is very slight, and does not vary. When wire 20 L.S.G. is used, and the tension on it is as much as it will bear with safety (about 200 lbs. is sufficient), the "sag" will not exceed 0·15 inch per 100 feet. At the sternpost and bulkhead holes, cross pieces of wood should be fixed and the centre transferred to them; with these centres circles should be "scribed" in and marked with a centre punch, when they serve as guides in boring out for the stern tube. It is sometimes found advantageous to verify the centre line markings by the system called "sighting"; this is done by placing battens horizontally at convenient places, whose upper edges touch the centre line of shafting, and, when viewed, should all coincide if they are exactly in line. A similar system of battens should then be placed vertically with the same result, if the line is straight. This plan, however, is a somewhat tedious one, and by no means reliable in most instances owing to the deceptive nature of the light in the hold of a ship.

Boring the Sternpost.—A boring bar with tool head, &c., is fixed accurately in position by means of the reference circles before-mentioned, and the sternpost, which has been roughly bored to within about 5 per cent. of the finished size before being fixed in place, is bored out to the exact size required; the bulkhead, with its liner, is also bored out, and also any other part into which the stern tube is required to fit accurately. The finishing cut through the sternpost should be commenced *from the inside*, as the wearing away of the cutting edge of the tool causes the hole to be slightly taper, and this allows the tube to be made a very tight fit in it.

Engine Seatings.—The superstructure raised on the ship's frames to carry the engines is called by various names, such as *engine bed*, *engine seatings*, *engine foundation*, *engine bearers*, &c., and is one requiring some skill to design and care to manufacture properly. As the success and efficient working of an engine very materially depend on this structure, too much care cannot be devoted to its construction. The weight of a marine engine is considerable and concentrated on a comparatively small area; in a seaway the inertia causes very severe stresses on the seating and on the bolts connecting the engine to it; and, in some cases, the strain of the engine itself when at work is borne largely by the bed on which it rests, owing to the want of rigidity in the bedplate.

The ship cannot always be viewed as a rigid structure, for elasticity is observable in all ships, especially when unloaded, and is very marked in those built for shallow water navigation of considerable length. For this reason, the engine seating must be so designed as to add materially to the stiffness of the ship's structure, and be of sufficient strength to transmit any stresses caused by the weight of the engines to the main framework of the ship. To this end, the vertical portions of the engine seating should be worked in or be in one with the floor plates and keelsons, and the longitudinals should extend beyond the immediate vicinity of the engine bed, so as to distribute the strain over a longer portion of the ship, and not localise it on a few frames. The longitudinal strength thus added to the ship's bottom should not end abruptly at the bulkhead, as is commonly the practice, but be continued beyond it and decreased gradually. The effect of stopping the engine seating at the aft bulkhead of the engine-room, is to cause a sudden change of flexure in the ship's bottom at that point, when the ship is steaming in a heavy sea; this change of flexure produces abnormal stresses on the shafting, especially on the after part of the crank-shaft; the after bearing of crank-shafts shows this by its tendency to heat, and, in extreme cases, the shaft is broken at the crank-arm, or at its junction with the crank-arm.

When the crank-shaft is connected to the thrust shaft by drivers, the working of the ship is proved by the squeaking sound emitted by them when not oiled.

The scantlings of the engine seating should never be less than those of the ship's bottom to which it is fixed, and when the seating is high, they should be in excess of the ship's scantlings. The angle-irons should be carefully fitted, and the riveting more than usually good; the rivet holes should be fair and well filled with the rivet; and if the holes are not fair, simply drifting them out to allow the rivet to pass through is not sufficient; these remarks particularly apply to the connection of the top plate with the verticals. The top plate should be at least 50 per cent. thicker than the vertical plates, and well bedded in place. Fifty per cent. of the rivet holes should be drilled through and through, and the remainder rimered fair and true to ensure a thorough trustworthy foundation for a modern engine.

It is the practice with some engineers to hold the engines down to the ship by a few large bolts which pass below the seating, and connect to strong cross-bars under the reverse frames of the ship; this is, however, not a good practice, as the strain is localised to an unnecessary degree, and such bolts are very apt to corrode rapidly from the action of bilge-water, and, being unseen, to break without being discovered.

The engine seatings are peculiarly liable to decay from the action of bilge-water and its gases acting on the warm metal; to prevent this they should be carefully protected by cement where practicable, and well painted where cement cannot be got to stick; cement is, however, better than paint, and if mixed hot, and washed on, will form a very efficient covering.

Thrust Block Seating.—This also, from its importance and the nature and the magnitude of the strains on it, must be carefully constructed. There should be three vertical plates, extending over, at least, four frames in small ships, and six to eight frames in large ones; the centre plate should be above, and strongly secured to the keelson by angle-irons, its thickness should be 50 per cent. more than that of the floor plates; the side plates should be 25 per cent. thicker than the floor plates, and connected to the reverse frames by strong angle-irons; all three verticals should be caused to abut on the engine seating and tied to it. The top plate should be of the same thickness as that of the engine seating, and when possible in line with it. Stop plates should be riveted to the top plate to serve for the thrust block to abut on. All the rivet holes in this top plate should be drilled quite fair with the holes in the angle-irons and connections, and care should be taken that all the holes in the thrust seating should be rimered fair and all rivets quite fill the holes. In ships of very large power, the base of the thrust block seating should extend over more frames, and the vertical plates should be worked intercostal with the floors, so as to form a direct tie to the ship's bottom plating.

Pedestals for Tunnel Shafting.—The plummer blocks for the tunnel shafts rest on the tunnel bottom when the shafting is not high, but when the distance is too great for this, pedestals are built of plates, whose thickness is about the same as that of the floors, connected by angle-iron, so as to form a stiff column; the top plate should be 50 per cent. thicker.

Boiler Seatings or Bearers.—The boiler, with its fittings and mountings, together with the water it contains, requires a very strong support and efficient means of keeping it in place when the ship is rolling or pitching.

When the boilers are placed athwartships (that is, their axes are athwartship), the bearers act as beams to distribute their weight over a large number of frames, and may be made of H section, so that the lower flange is riveted to the reverse frames, and the top flange carries the chocks, which are wedge-shaped, and shaped to fit under the boilers, and form saddles for them to sit in. Single-ended boilers should have two such saddles, whose breadth of face where the boiler rests should not be less than nine inches for very large boilers, and six inches for small ones. Double-ended boilers of moderate length and size may have three such saddles, but long or large double-ended boilers should have four sets. The chocks are sometimes made of angle-iron and plates, but they are better made of cast iron or cast steel; when of the latter materials, the patterns can be tried in place after the boilers are in position, and made in such a way that the chocks, when cast, will fit with sufficient accuracy as to require no packings; when made of wrought iron, much expense is incurred in trying to make them fit, and in the end packing is often necessitated.

If the boilers are placed "fore and aft," that is, with their axes longitudinally, the bearers are generally laid on the top of individual floors, thus localising the weight on a few frames only. To avoid the straining action proving dangerous, the longitudinals of the ship in wake of the boilers should be increased, and extra connections made between them and those frames carrying the bearers. Sometimes the boilers when in this position have been carried by longitudinal bearers inclined so as to be in planes passing through the axis of the boiler. Such bearers distribute the strain over a considerable number of frames, but do not so well support the boiler, and moreover prevent access to the boiler bottom for examination and repair.

Fitting Machinery on Board the Ship.—The stern tube and the screw shaft are fitted into place, and all sea-cocks and valves fixed to the skin of the ship before it is launched. After it is in the water, the tunnel shafting is placed in position piece by piece, each one being set so that its coupling comes fair and true with that of the preceding one; the shaft bearings are raised on temporary packings until the whole of the shafting is in place and coupled up; when this is done, the shafting should be turned around so that the bearings, if not placed in *exact* position at first, may adjust themselves; the bearings should now have the proper packings fitted to them, and when bolted down, the coupling bolts should be withdrawn, and each shaft tried around to see that there is no want of correspondence at the couplings. This may seem a somewhat tedious process, but it is a very safe one, and one which prevents all possibility of shafting being fitted out of line; the engineers of the ship should occasionally withdraw the bolts, and prove the shafting true, especially after the ship has had cause for straining. Of course, the accuracy of this method depends on the care with which the couplings have been turned; but, as modern appliances are capable of turning a shaft coupling quite true with very ordinary care, there is little cause for fear on this ground.

The engine bed-plate, or foundation plate, with the crank-shaft in place, is now lowered on board, and placed on temporary packings of iron; it is, by means of jacks, brought to its exact position, and proved by the shaft couplings as before. Permanent packings of *cast iron* are now carefully fitted between the temporary ones and the latter withdrawn, and their places filled with hardwood (teak, greenheart, elm, mahogany, &c.) packings formed of pairs of wedge pieces driven from opposite sides.

The Holding-down Bolts should be carefully fitted so as to distribute the load over the bed, and means provided to prevent the nuts from slacking back; the Admiralty require check nuts to be fitted to all holding-down bolts, but it is better to prevent movement of the nut by slightly riveting the bolt end, than to trust to check nuts.

Staying Engines.—Vertical engines, especially those of long stroke, are supposed to require some means of support to prevent undue straining of the columns when the ship is rolling heavily. Such engines, when supported by *vertical wrought-iron* columns, do occasionally show signs of such a need; but it is better to bear this contingency in mind when designing the columns, and make them sufficiently strong and rigid to withstand such strains, than to trust to getting support from the elastic hull of the ship. In case of a collision, such supports, if rigid, might be a positive source of danger, as the force of the blow delivered in their neighbourhood might seriously damage the engines, if not destroy them altogether.

Such support as is sufficient can be provided by splaying out the front columns, as shown in fig. 12, and making them of strong cast iron or cast steel.

Ramming Chocks and Stays.—The Admiralty require all vertical engines to be fitted with non-rigid stays from the cylinders to some part of the ship in such a way as to keep the columns, &c., from being damaged or strained when ramming the enemy. Boilers also must have stays and chocks to keep them from being jerked out of place when ramming.

Boiler Seats.—The boilers require to be carefully fitted in their seats, and secured there so that they cannot be displaced by the rolling or colliding of the ship. The boiler should require no packings when the chocks are of cast iron; but when the saddles are of wrought iron, and do not fit the boiler exactly, it is better to interpose iron packings at intervals, and fill in the spaces between with hardwood wedge pieces. To prevent movement longitudinally, "toe" plates should be riveted to the frames or other convenient part of the ship's structure, and these should be stiffened by angle-irons. The "toe" plates should stand about 6 inches above the bottom of the boiler, and be clear of the man-holes and mountings. The boiler is held in its seating by straps surrounding it, and secured to the bearers when of small size; but the general practice is to secure it by tie-bars from its upper part to the side of the ship, or by struts formed of plates and angles from the stringers, bearers, &c. Each particular case requires special treatment, and it is impossible to lay down any rule, beyond that of providing for every contingency to which a ship is liable. Tie-bars from the boiler sides to the bearers is a good and perhaps the best way to secure them, and is the Admiralty practice.

At one time the Admiralty practice was to lay the boiler in a bed of mastic cement, spread on a cradle formed to suit the boiler bottom. This was very necessary for the box form of boiler, especially in wooden ships. This cement also practically insulated the boiler, a practice which may be followed now with advantage to both ship and boiler. A strip of asbestos, $\frac{1}{4}$ to $\frac{1}{2}$ inch thick, interposed between the boiler and its saddle, is a good arrangement for preventing thermo-electric action on the ship's frames, tank top, &c., as well as for retaining much heat in the boiler. The boiler bottom in iron ships should be well above the bilges, and room provided for a man to get in to paint or repair; the boiler bottom and ship's framework can then, and should, be kept well painted. The boiler bottom should also have a good coat of non-conduction material applied to it in such a manner as to prevent radiation, and at the same time such as to permit of easy removal and replacement for examination and repairs to the shaft.

Copper Pipes.—The whole of the pipes subject to internal pressure should be of steel, wrought iron, or strong copper; the exhaust pipes may be, and usually are in the mercantile marine, of cast iron. The Admiralty require that all pipes of steel or copper up to 6 inches diameter shall be *solid drawn*. Wrought-iron and steel pipes are now very generally employed, especially for large sizes and high pressure. If of steel the Board of Trade require a butt strap to be riveted over the weld.

The thickness of the main steam pipe should when of copper

$$= 0{\cdot}125 + (\text{diameter of bore} \times \text{pressure} \div 10{,}000).$$

The thickness of feed pipes when of copper

$$= 0{\cdot}125 + (\text{diameter of bore} \times \text{pressure} \div 8000).$$

The thickness of blow-off and scum pipes

$$= 0{\cdot}125 + (\text{diameter of bore} \times \text{pressure} \div 9000).$$

When made of steel or iron welded the thickness should be 20 per cent. more than given by the above.

The thickness of copper or brass main inlet pipes

$$= 0{\cdot}1 + (\text{diameter} \div 300).$$

The thickness of copper or brass main discharge pipes from reciprocating pump $= 0{\cdot}1 + (\text{diameter} + 200)$.

If for a centrifugal pump the discharge may be of the same thickness as the inlet pipes.

The thickness of copper feed suction pipes and bilge-discharge pipes

$$= 0{\cdot}09 + (\text{diameter} \div 200).$$

The thickness of copper waste steam pipes

$$= 0{\cdot}05 + (\text{diameter} \div 500).$$

The flanges for copper or brass pipes should always be of tough brass, and of a thickness equal to 4 times that of the pipe; the breadth of flange should be 2¼ times the diameter of the bolts used. For pipes exposed to a pressure of 30 lbs. and upwards, the pitch of the bolts should not exceed 5 times their diameter, or 5 times the thickness of flange; their diameter is usually about the same as the thickness of flange. For pipes not subject to a pressure of more than 30 lbs., the bolts may be 6 diameters apart, or even a little more in some cases. All pipe flanges should be machined "true," especially those of the feed and steam pipes. These should fit face to face without forcing, and be jointed with as little putty or paint as possible. Brass flanges on the copper pipes may with advantage have grooves into which is fitted a ring of copper wire soft enough to easily conform to the grooves when pressed by the bolts. This can also be done with iron flanges, but the Admiralty complain that severe corrosion has taken place with them so done. Corrugated rings of soft copper or brass have also proved good jointers when fitted inside the line of bolt holes. The Admiralty require the whole range of piping to be tested by water and then by steam. The jointing that does best for water is not generally so good for resisting steam and heat. The bolting should be close, rather than wider apart with larger bolts.

Starting and Reversing Engines.—It is most important that an engine shall be capable of having its motion instantly reversed; in fact, no engine is satisfactory which cannot be stopped and made to move full speed astern in less than 30 seconds from the time of the engineer commencing the evolution. This can scarcely be done by the hand-gear with even comparatively small engines, and is beyond possibility with large ones. It is essential, therefore, to provide mechanical means for this purpose in all engines of over 100 N.H.P., if such efficiency is required; and it should be of such power as to perform the operation without shutting off steam. The simplest method is to fit a steam cylinder with rods, &c., to a lever on the weigh shaft in addition to the ordinary hand-gear, the piston pushing and pulling as required to help the engineer. There is, however, the objection to this, that the cylinder is then somewhat large, and at times the steam-power masters the hand-power, and overruns its limit, thereby tending to cause damage. This latter objection, however, is easily got over in many ways, the best of which is by adding a second small cylinder containing water or oil, which is forced by its piston from one end to the other, and thereby acts as a brake to the gear.

Brown's Patent Reversing Gear.—This idea of the brake cylinder has been worked out, and perfected by Messrs. Brown of Leith, who make a gear which, by the movement of a small lever easily moved by one hand, operates on the valve motion instantly, and only to the exact extent intended by the operator, so that if the engineer moves the lever through one-quarter of its angular movement, the link-motion is moved by the gear through exactly one-quarter of its traverse. This is effected by means of a system of levers, so arranged that the gear, by moving, replaces the valves in the exact position from which they were displaced by the hand-lever. Messrs. Brown have also devised an arrangement whereby the hunting and compensating levers with their rods are on the engine and so the apparatus is self-contained. Figs. 188, 188*a*, and 188*b*, show the methods adopted by the firm to make the gear automatic.

Steam Gear for Reversing.—The simplest and most efficient of the steam gears, and one whose cost is so small that it may be fitted to the cheapest of engines, consists of a small engine, on whose crank-shaft is a worm, which works in a worm-wheel capable of turning freely on a fixed gudgeon on the engine frame or other convenient place; on this worm-wheel is a stud or crank-pin, to which is fitted a rod connecting it to a lever on the weigh-shaft; the eccentricity of the crank-pin is equal to half the chord of the arc through which the lever end works. The engine being set in motion, the worm-wheel is caused to revolve, and the motion of the crank-pin causes the weigh-shaft to oscillate, and so to reverse the links. The steam-cylinder is sometimes fitted with reversing-gear, but it is quite unnecessary, and is really better without it, as the little engine moves so fast that practically no time is lost in making a complete revolution of the worm-wheel.

The advantage of this gear over others, besides its cheapness, is its simplicity, safety, and capability of being used to turn the engines when in port, or to work a winch for lifting weights when overhauling the engines; the hand-wheel is also in this case on the little engine shaft, and acts as a

flywheel. The Admiralty engines have a similar starting-gear, but fitted with two cylinders whose cranks are at right angles; the starting hand-wheel is only connected to the crank-shaft or gear when required to be worked by hand; this permits of the engine starting from any position without help and stopping at any point without a brake, as is often necessary in the mercantile marine (fig. 12).

Brown's Patent Steam Starting Gears.

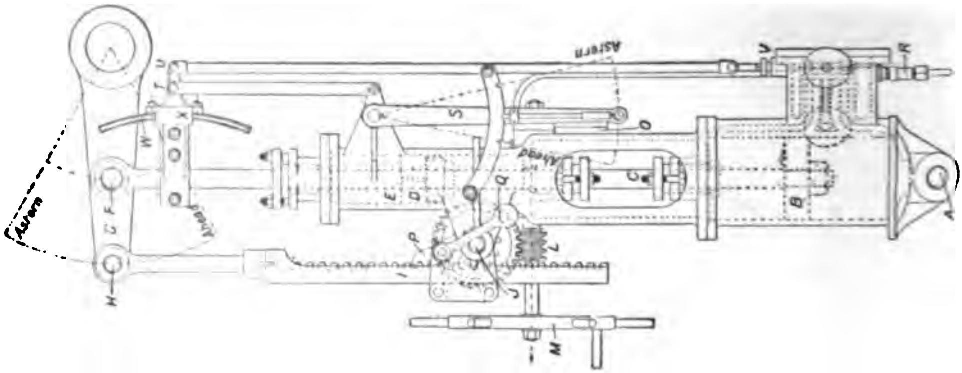

Fig. 188b.—Latest Design.

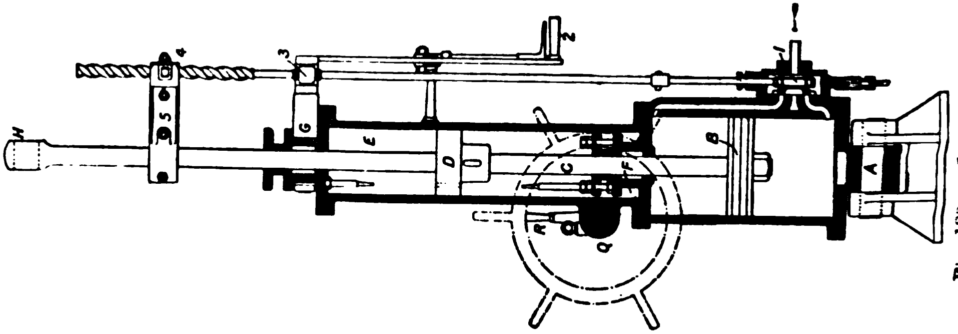

Fig. 188a.—Improved with Ordinary Hand-Gear.

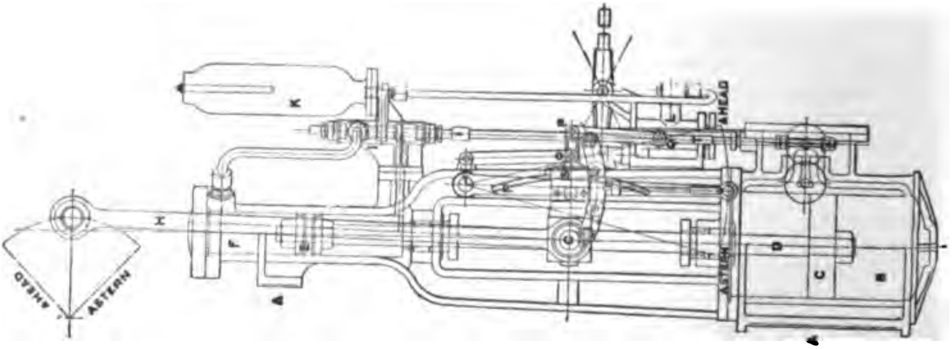

Fig. 188.—Original Plan with manual control by a pump.

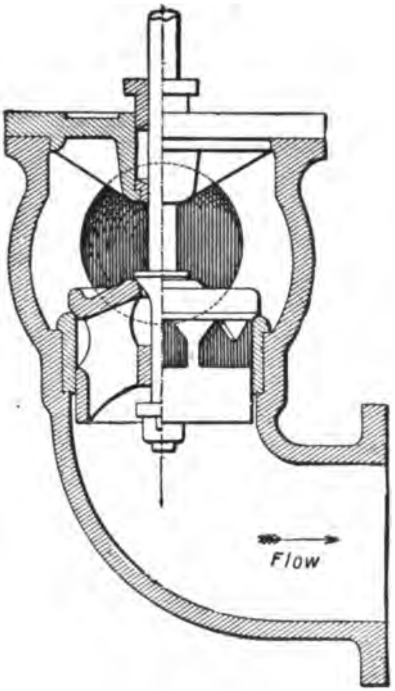

Fig. 189.—Engine Stop and Regulating Valve (for small engines).

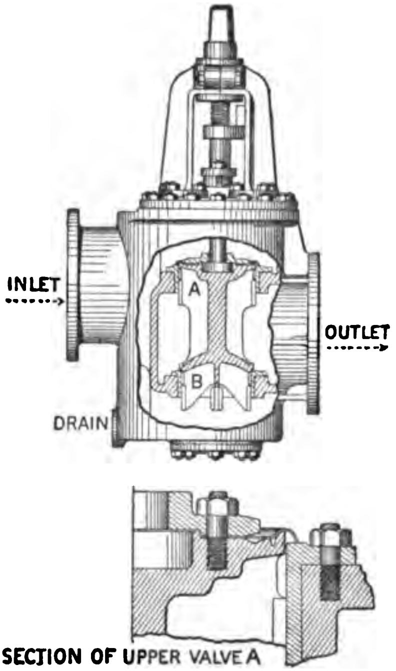

Fig. 190a.—Equilibrium Stop and Regulating Valve (Bevis & Gibson's Patent).

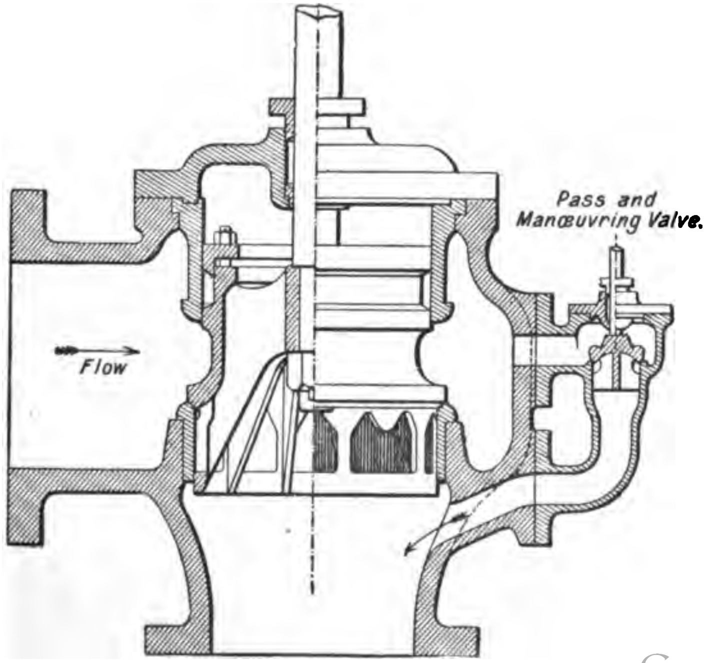

Fig. 190.—Balanced Stop and Regulating Valve (for large engines).

Regulating and Stop Valves.—As it is necessary that a marine engine may run at any number of revolutions from full speed to dead slow, means must be provided for doing this efficiently, especially for war ships whose speed must be regulated to a nicety. In the old days speed was decreased by means of a throttle valve, or by "notching" up the link motion. The latter method is still adopted with advantage, but it is not sufficient when very low speeds or minute changes in speed are necessary. The old Butterfly throttle valve is now so disused, except for governing purposes, and against the high pressures now obtaining it is never tight enough for practical purposes.

Fig. 189 shows a form of stop regulating valve which has been used successfully in the mercantile marine for many years. The little valve on the spindle is easily opened, and is sufficiently large to supply the necessary steam for manœuvring purposes if it is about one-third the diameter of the main valve. Large valves of this kind offer considerable resistance to opening, notwithstanding having a pass valve, and there are other objections, so that many engineers prefer to have some form of balance stop valve, such as is shown in Fig. 190, with an independent manœuvring valve on the side.

Fig. 190*a* is a form of equilibrium stop valve, self-adjusting, so that it keeps tight under differing pressures and temperatures.

The regulating valves, when properly balanced, can be worked by a lever, such as was used in the throttle valve, or with a wheel at the starting platform in the ordinary way. Very large valves, however, are too cumbersome to be easily and quickly worked by hand, and may, therefore, have an apparatus something like the Brown steam starting gear attached to them, and operated by a small lever at the starting platform.

Steam Turning Gear.—Another labour-saving appliance now universal in the Navy and mercantile marine, is steam gear for turning the engines when in port. In all but small ships a separate engine is provided for the purpose, but in small ships the donkey pump engine, or the reversing engine, when there is one, is employed by using belt or rope gear. The usual plan is to fit a second worm-wheel to the worm-shaft, and to turn it by a worm on the shaft of the special engine, or on a shaft with a pulley to be worked by an auxiliary engine.

Steam Ash Hoists.—All large ships require some mechanical means of disposing of the ashes, clinker, &c.; the simplest form of hoist is only a small winch worked by a steam engine without wheel-gearing; the barrel is of small diameter, and the flywheel is heavy, and kept running at a constant speed, the bucket of ashes being "whipped" up in the same way that light cargo is got out of the holds with a steam winch. This gear has the merit of cheapness and simplicity, it can be worked by the most ignorant, and does not easily get out of order.

Ash Hoist (See's), fig. 191, is an ingenious arrangement whereby the ashes are driven to the top of the tube, tilted over, and shot through the ship's side. With this gear, the fireman is not required to leave the stokehole, and no labour is required beyond that of shovelling the ashes into the hopper. Fig. 191 shows the arrangement to consist of a hopper having a hinged water-tight cover; at its bottom is a special form of nozzle, which discharges water up the delivery pipe from a special duplex pump, when the pressure has been got up in it to 200 lbs. As soon as the gauge shows the pressure, the cock is quickly opened and a rush of water up the pipe takes place, carrying with it the ashes in the hopper; the pressure drops to 150 lbs., but the stream still flows through the pipe carrying air and ashes with it as fast as the latter are shovelled into the hopper. When the whole are disposed of, the cock is closed *quickly* to prevent back wash; the hopper is closed, as is also the discharge valve on the ship's side. This apparatus is a great convenience and comfort, especially in bad weather, and saves a huge amount of unpleasant labour; it also prevents the need of the grimy and unwashed denizens of the stokehole showing themselves on the decks of yachts or passenger vessels.

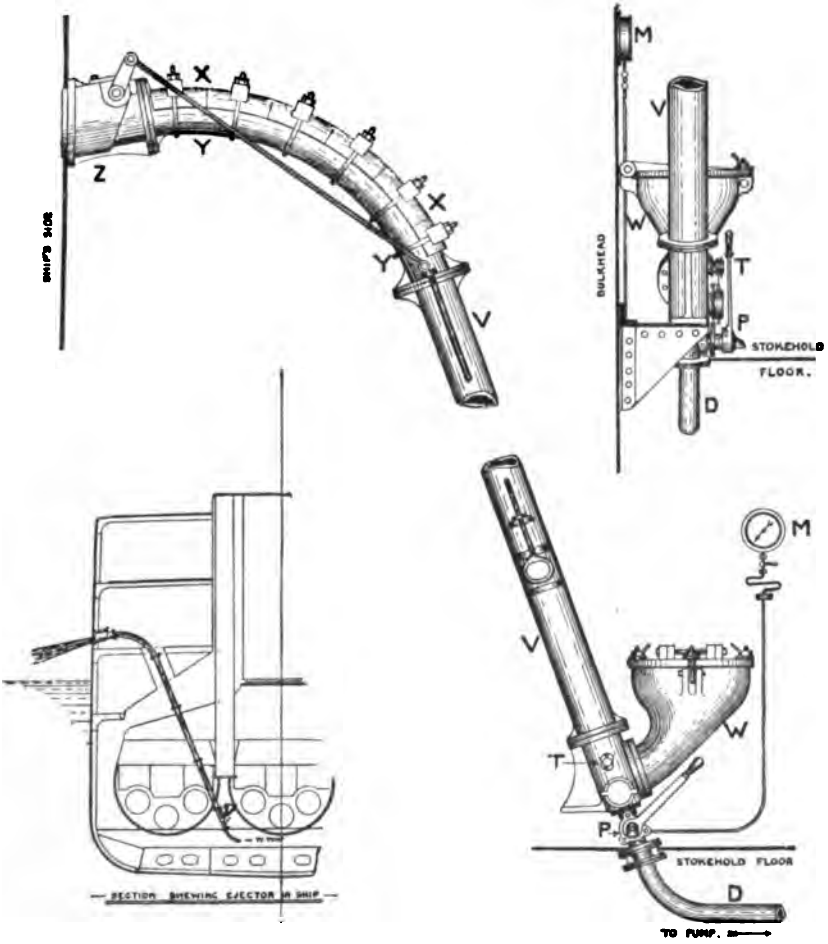

Fig. 191.—Ash Ejector.

Governors.—To prevent the engines from racing when the sea is rough, it was usual to fit an instrument which should control the throttle-valve and work automatically. The governor, as fitted to land engines, is not admissible on board ship, owing to its susceptibility to derangement from the motion of the ship.

There are two distinct classes of marine governor—viz., those whose action is influenced by the variation of motion of the engine itself, and those whose action is caused by the variation of pressure at the stern, due to variation of head of water. The former class can only act *after* a variation of speed has taken place, the latter anticipates and checks such variations. At first sight the latter present the most favourable qualities, inasmuch as they *anticipate* change of velocity, but they serve only one purpose, that of checking *racing* of the engine due to the propeller emerging from the water. The other governors are necessarily a little late in action, but they may be made so sensitive as to be almost as quick as the others; they have, however, one superior merit, and that is, they check racing from any and every cause. If a propeller or shaft break, the pneumatic or hydrostatic governor fails to check the engine; the other governors will check any large increase in velocity and give the engineer time to shut off steam.

Both classes of governor are susceptible of subdivision, and may be distinguished as those which act direct on the throttle-valve, and those which act on the valve of a steam cylinder whose piston operates on the throttle-valve.

Silver's Governor.—The principle of this governor is to obtain a motion for its gear, which operates on the throttle-valve, by means of the variation in the velocity of a heavy flywheel and its shaft, which latter is driven by the engine, and on which it may revolve loosely; the wheel is capable of only small angular movement with respect to its shaft, and is set in motion by the "stops" acting on it; its motion is kept sensibly uniform by its inertia, and by vanes attached to it acting on the air. The motion of the gearing operating on the throttle-valve is obtained by a bevel pinion on *the wheel*, acting on two bevel quadrants working on gudgeons on the *shaft;* so long as the shaft and wheel work at the same speed there is no motion of the quadrants; when there is a change of velocity the quadrants turn on their axes, and the throttle-valve is moved.

Merlton's Governor.—This works on the same principle as Silver's, but has different gearing to obtain the longitudinal movement of the part which actuates the throttle-valve. Here the heavy wheel has cast with it parts of two helices, which fit two corresponding parts on a coupling; it is drawn round with, and free to slide on, the governor shaft; the wheel and coupling are pressed together by a coach spring.

There are some other governors which have been used for marine purposes, but as they are now entirely superseded by steam governors, it is unnecessary to particularise them.

Dunlop's Governor.—This instrument consists of a vertical cylinder placed close to the stern of the ship, and as low down below the water line as convenient; there is a communication between the sea and the bottom of this cylinder by a cock or valve of ample size, fitted to the skin of the ship. When the stern of the ship is lifted out of the water, the cylinder is emptied of water, and the pressure in it is that of the atmosphere; on the stern dipping deep into the water again, the water rushes into the cylinder and compresses the air in it till there is a pressure due to the "head" of water, which may amount to as much as 12 lbs. per square inch in large ships, and 5 lbs. even in small ones. The top of the cylinder communicates by means of a pipe with a vessel in the engine-room, which is closed air-tight at its top by a thin corrugated circular diaphragm. The bottom of this vessel is bell-shaped, so that at its top it is of considerable diameter, and the diaphragm capable of exerting considerable force when even so small a pressure as 1 lb. per square inch is transmitted from the cylinder. The gear for operating on the throttle-valve is connected to the middle of the diaphragm, and any bulging of it by the pressure causes the throttle-valve to be opened, and when the water cylinder is emptied by the rising of the ship's stern relatively to the sea, the diaphragm assumes its normal position, and the throttle-valve is closed.

Smith and Pinkney's Governor differs from both classes mentioned, inasmuch as it is neither actuated by the motion of the engine nor by the pressure of the water. It consists of a heavy pendulum set so as to oscillate in a fore and aft vertical plane; it will consequently have relative motion when the ship pitches; the pendulum is connected to the valve of a small cylinder, whose exhaust communicates with the condenser, and whose valve-*box is open to the air* (with the mistaken notion that "vacuum costs nothing"); the piston of this small cylinder is connected to the throttle-valve, and is sufficiently large to easily control it even when the spindle-gland is packed tightly.

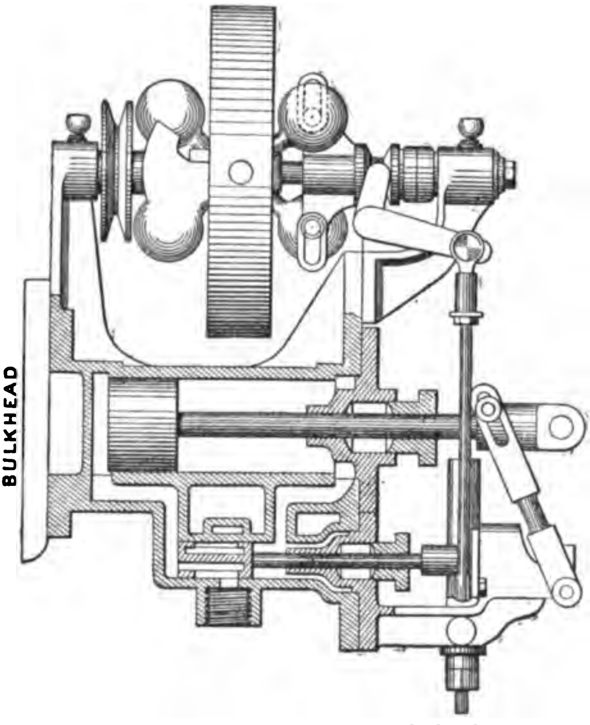

Fig. 192.—Combination Governor (Murdoch's Patent).

The objection to this kind of governor, in addition to the one urged against all governors not actuated by the engine shaft, is that, with a "following" sea, the ship may be depressed at the bows and have the propeller fully immersed; in a short "choppy" sea, the propeller may be bare, and the ship not "down by the bow" at all, so that the governor would not check the engine from "racing."

In practice, however, it is very quick in operation, and under the ordinary circumstances of a "head" sea answered the purpose of preventing racing admirably. It is arranged as a *pneumatic* governor, but it would be better if steam were used instead of air; there is no saving in using the latter, and it is found in practice to seriously interfere with the vacuum when the ship is exposed to a "head" sea, and tends to charge the feed-water with air to the detriment of the boiler plates, &c.

Steam Governors.—The chief cause of failure of the first marine governors was their inability to move the throttle-valve promptly when the spindle-gland was tightly packed; it was also difficult to set them so as to act efficiently when they would move at all. The difficulty was got over by limiting the function of the governor to move the slide-valve of a small steam cylinder whose piston performs the operation of opening and shutting the throttle-valve. The governor is of ample power to move the small slide-valve with precision, and the steam cylinder can always be made of sufficient size to work the throttle-valve however tightly its gland is packed. For the purpose of working the small slide-valve, a Silver's or a Meriton's governor may be employed, and many of the old governors of this kind were converted to steam ones by the addition of this steam cylinder.

Durham's and Churchill's Velometer.—The motion of the engine is communicated to this governor by a small rope of wire or Manilla on the usual pulleys; it is transmitted from the pulley-shaft to the shaft of a paddle-wheel enclosed in a cylindrical trough, by means of a bevel-wheel, whose axle is free to move about the axis of the shafts in a plane perpendicular to it, and which gears into a similar bevel-wheel on the end of each shaft. (This is similar to the arrangement provided in traction engines to admit of their going around a curve.)

The trough is filled with water or oil, which is carried around with the paddle-wheel, and causes it to resist any sudden changes of motion. The axle of the intermediate bevel-wheel is connected to the small valve of the steam cylinder whose piston operates on the throttle-valve. If the engine races so that the bevel-wheel on the pulley-shaft moves faster than that on the paddle-wheel axle, it will carry the intermediate pinion along with it, until the motion of the paddle-wheel is accelerated by it, and the pinion axle acting on the small slide-valve causes the throttle-valve to be closed. The paddle-wheel is now moving at a higher rate than its normal speed, so that when the engine has slowed down to its normal speed, the bevel-wheel axle is moved in the opposite direction, so as to cause the throttle-valve to be opened, and the engine is thus prevented from further "slowing down."

This governor, when carefully adjusted, is most sensitive, and will prevent any dangerous racing under the most trying circumstances.

Coutt's and Adamson's Governor.—This is an extension of Dunlop's principle, and differs from it by the diaphragm being caused to move the slide-valve of a small steam cylinder whose piston operates on the throttle-valve. This has the advantages and disadvantages of Dunlop's governor, except that the work of moving the throttle-valve is done by steam.

Westinghouse Governor.—A sensitive ball governor, of very small size, is made to operate on a valve which, in opening, allows the steam on one side of the piston of a steam cylinder to escape to the condenser, that on the other side forcing the piston to move quickly and shut the throttle-valve; the valve closes again as soon as the steam has escaped, and the steam flowing into the cylinder allows the piston to go back to its original position.

The piston here spoken of is connected to a smaller piston in another cylinder, whose function is to bring it back to its initial position.

This governor is very sensitive and acts very well, but is liable, from want of attention, to get out of order, and then fail to act at all.

With the advent of triple engines and twin screws the absolute necessity for a governor disappeared; and to-day, although some racing takes place with all ocean-going marine engines, hardly a single modern ship is furnished with a governor, and when such an instrument has been provided, it generally supplies ample evidence of its non-use if not of its uselessness.

Gauges.—In every engine-room there should be a steam gauge, which shall show the pressure *at the high-pressure cylinder valve-box;* a gauge which shall show the pressure in each receiver (when the engine is compound), and that on the low-pressure is called a *compound* gauge, as it is marked as a *pressure* gauge when above atmospheric pressure, and as a *vacuum* gauge when below—this gauge might, with advantage, be marked so as to show *absolute* pressure; if this were so an additional stumbling-block would be removed from the path of our "practical" engineers, and allow some of them to have less hazy views on the question of "vacuum"; and a gauge, commonly called the vacuum gauge, which shall show the pressure in the condenser. This gauge is marked in inches, and so indicates how high the column of mercury would be in a vertical tube whose upper end is connected with the condenser, and the lower open to the atmosphere. The old original vacuum gauge was simply like a barometer, and, when replaced by the Bourdon gauge, to avoid confusion the new ones were marked in this way. To say that there is a vacuum of 12 inches, means that the difference between the pressure in the condenser and that of the atmosphere is equal to the weight of 12 cubic inches of mercury, or due to a "head" of 12 inches of mercury, or 6 lbs. very nearly.

It would be far simpler, and certainly more scientific, to have the condenser gauge marked from 0 to 15 lbs. *absolute;* the "compound" gauge, or that attached to the valve-box of the low-pressure cylinder, marked from 0 to 50 lbs. *absolute;* and that attached to the valve-box of the other cylinders to 200 lbs. *absolute*, or to such limit as shall be at least 25 per cent. higher than the working pressure.

The gauges on the boilers might be marked as at present—viz., to indicate the pressure above that of the atmosphere, as it might be more difficult to train firemen to know the meaning of the new markings, but after all it is really immaterial to them how it is graduated, so long as they know to what mark they must keep the pointer when under steam, and that, when the pointer begins to move, on getting up steam, pressure is forming.

Care should be taken in setting the gauges in the engine-room that allowance is made for the extra pressure due to the "head" of water in the gauge pipe, which will be about 1 lb. for every 2 feet of vertical fall.

Lubricators and Impermeators.—To obtain perfect lubrication the supply should be steady, uniform, and continuous, or nearly so. This is true for every bearing, guide, &c., and equally true for the lubrication of the internal parts. It is usual to rely on capillary attraction to convey the oil from the oil boxes to bearings, guides, &c., by means of worsted syphons; this is a very simple, well tried, and fairly efficient method, but it has serious drawbacks; it requires constant attention, as the worsted wick syphons act as filters and become clogged with gluey matter contained in some oils, and there is no definite means of *proving* that the oil is passing. An equally simple and very satisfactory plan (fig. 193) is to fit an oil box a few inches above each bearing, in such a way that if oil is dropping from it, it can be seen or felt; there is a small cup to each oil hole leading to the bearing, and over each of these is a small nozzle from the bottom of the oil box, fitted with a small plug so as to regulate the flow of oil, or to stop it altogether; if preferred, however, syphons may be fitted instead of screw plugs, as in either case the *flow of oil can be proved.* A perfect lubrication, however, can only be got by forcing the oil by means of a pump through every joint and bearing, collecting the oil as it drops, cooling it, and again pumping it to the joints. This system of forced lubrication introduced by Messrs. Belliss & Morcom, and applied to their high-speed electric light and power engines, has proved a great success by permitting of a mechanical efficiency of 93 per cent., and preventing wear of bearings and journals during quite long periods of work.

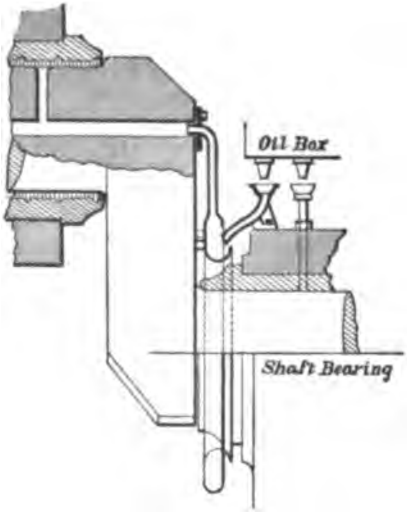

Fig. 193.—Centrifugal Lubricator.

Cadman's Patent Lubricators.—*Over* each moving part required to be lubricated is an oil box having, projecting through the bottom, a small plug, held there by a spring, and so set that the oil box on the moving part touches the plug end and opens it so as to let a drop of oil pass. This is especially adapted for the piston and connecting-rod brasses of a vertical engine, and for guides, &c.

Centrifugal Lubricators.—When engines are running at a high speed, ordinary external appliances, such as telescopic pipes, jointed pipes, pipes fixed to the rod and taking oil from wipers, &c., are not reliable or sufficient for the purposes of thoroughly lubricating the crank-pin brasses. If a system of forced lubrication by means of holes in the crank-shaft, its arm and pin, is not adopted, it is usual to have what is known as a centrifugal lubricator, such as is shown in fig. 193, whereby the oil supplied from a fixed lubricator passes through a tube, drops into a circular collar surrounding the shaft, and is whirled into its outer part and from it by means of a tube to a hole through the axis of the crank-pin, from which it flows by smaller holes to the surface and so lubricates the brasses. From the time of leaving the fixed lubricator centrifugal force is operating to compel the oil to flow to the brasses. Lengthened experience with this in the Navy has proved its reliability, and, as it is not an expensive fitting, it might be adopted with advantage generally in the mercantile marine, as less oil is wasted by it than by the ordinary appliances. The hole in the pin also provides a store of oil which would keep the brasses supplied in case of a temporary stoppage of supply from the oil-box.

Sight Feed Lubricator (fig. 194) is for the purpose of supplying a steady and definite supply of oil to the cylinders, and is so designed that the condensed steam in a pipe, &c., leading from the steam pipe, shall carry with it so many drops or globules of oil as it drains back near to the valve-box; the water condensed passes through a glass tube, near to the bottom of which the oil enters it, and the number of oil globules passing per minute may be counted, and the supply thus regulated to a nicety. This is a most ingenious contrivance, and in the hands of careful men who understand it effects great economy, while adding to the efficiency of the engine; but in careless hands it is almost useless.

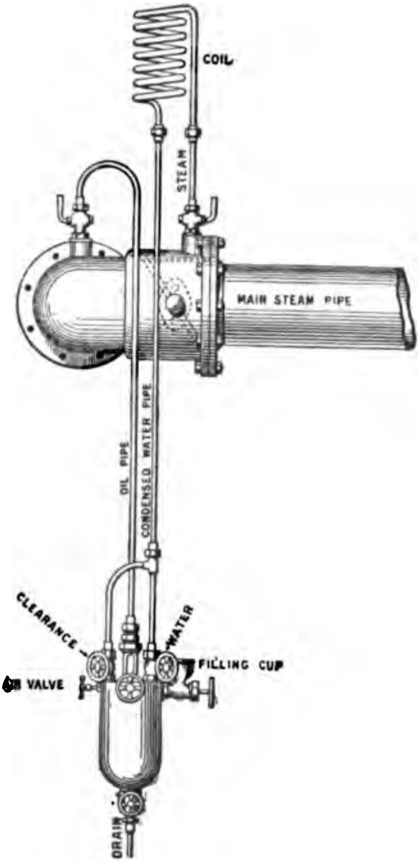

Fig. 194.

Mechanical Impermeators.—These are most successful in operation, and more reliable than any other form, inasmuch as their action is so simple that every one can understand them. Essentially there is only a force pump, having a very slow motion imparted to its ram or rams by gearing, moved by one of the working parts of the engine. The gearing generally consists of a ratchet lever worked from the valve-rod of the nearest cylinder, and moving a wheel on whose axle is a worm which gears into a wheel on the rim of a nut; this nut fits on the thread cut on the ram, and held in position by a bracket; as the nut is moved round, the arm moves slowly in or out of the chamber. The ram chamber is supplied with oil from a small tank, and is connected by a pipe to the top of the valve-box of the high-pressure cylinder; small non-return valves are fitted, so that the oil cannot flow back to the tank, and the steam cannot force itself or the oil back into the chamber. It is, however, a very common practice now to run engines without internal lubrication, or with only such as enters the cylinders with the piston and valve-rods. It has been found that after an engine has worn its internal moving parts fairly smooth there is no need of other lubricant than the moisture from the steam, and so all risk of damage to boilers from grease is avoided by doing away with internal lubricating apparatus. If, however, superheated steam is used, oil lubrication may become necessary in some engines; when it does, care should be taken to use only a pure hydrocarbon suitable to the temperature or graphite paste forced in by a similar instrument.

Drain Pipes from the cylinders and valve-boxes should lead to the condenser, so that there shall be no loss of fresh water, and no filling of the engine-room with vapour when the cocks are opened. It is customary to connect these pipes to the hot-well, which serves this purpose very well with high-pressure and medium-pressure engines; but since the pressure in the low-pressure cylinder of a compound engine is only for a very small portion of the stroke above that of the atmosphere, the opening of the cocks will not get rid of the water, but only allow air to force its way back, and so reduce the vacuum. It is sometimes convenient to see if water is flowing, and so prove that the cocks are not choked; this may be accomplished by fitting a "three-way" cock to the main drain pipe, which permits communication to be made with the condenser or bilge at the will of the engineer.

Jacket Drains should always lead to the hot-well, and when the engine is working the cocks should be open sufficiently wide to just keep the jackets free of water; the hot water escaping from the jackets then helps to warm the feed-water.

Feed-heaters.—It is a most essential thing that the feed-water shall enter the boiler as warm as possible, both to effect economy of fuel and avoid wear of the boiler. Economy can only be effected *directly* by making use of heat that would otherwise be wasted for this purpose; but *indirectly*, by promoting circulation and reducing the necessity of circulation instead of checking it, considerable economy may be effected, even to the extent of warranting the use of heat which is not "waste." Many attempts have been made to heat the feed-water with exhaust steam, hot gases in the uptake, &c.; but no great measure of success has attended the efforts of those who have paid most attention to this, for the apparatus employed has generally been inefficient and its durability short. In the older expansive engines, where the temperature of the steam at exhaust was often over 230° Fah., great economy was effected by heating the feed-water in a small kind of surface-condenser, placed on top of the condenser so as to intercept the hot current of steam flowing to the latter; but now with compound engines, where the temperature of exhaust is only about 180° Fah. at the most, no such means is efficient for the purpose. No doubt *some* economy is possible even under these circumstances, especially if the feed-water is permitted to *circulate* in the heater for an appreciable time; but considering that when it leaves the hot-well it should and can be at a temperature of 130° Fah., very little more heat can be imparted to it from an external heating agent whose temperature is only 50° above it. Some day, perhaps, means will be found to avoid the loss of all the *latent* heat which takes place, and which is huge compared with the whole of the *sensible* heat. The exhaust steam being at 180° Fah., the total possible saving of sensible heat is now 50° Fah., while the latent heat lost is nearly 1000° Fah.

Weir's Feed-heater is designed to raise the temperature of the feed-water to nearly 212° Fah. by means of a portion of the steam from the receiver of the compound engine.

Weir's Feed-Water Heater and Automatic Regulating Gear.—Fig. 195 gives a sectional view showing its general construction. The heating steam is taken from the low-pressure casing of the main engine and the exhaust of the auxiliary engines, such as feed pumps, electric light, fan engines, &c., is also led into the heater through the non-return valve, B, on the side of the apparatus. A circular ring and conical spray piece with perforations are fitted to mix the water and steam uniformly. The feed-water is forced by the main engine feed pumps through the spring loaded valve, D, on the cover in a thin sheet, and is instantly heated by contact with the steam. As the pressure in the heater is generally much less than that of the entering water, the effect of this lowering of the pressure, and sudden heating of the water, is to liberate the air in the water, and this is removed to the condenser, or to the atmosphere, by a small cock, K, on the air vessel placed on the top of the heater. The feed-water is thus rendered non-corrosive, and falls to the bottom of the heater at the boiling temperature due to the pressure. The steam admission valve is of special construction. It can be opened to admit the necessary amount of steam to the heater, but it closes by its own weight in case there is no flow of steam into the heater; this valve is also fitted with a dashpot, which allows the valve to close gradually, and prevents it hammering on its seat in case of fluctuations of pressure. The combination of the automatic regulating gear with the heater has long been a special feature of the Weir apparatus. The float, E, shown in the lower part of the heater is a pan, with water-tight bottom and sides, but open on the top. It is suspended on two levers, so as to move up and down with a parallel motion; the top lever spindle is carried through the door at one end, and is balanced by a lever and weight. The float is always full of water, and the weight is adjusted to balance when one half is immersed in water. To the weight lever another lever is attached, which actuates the throttle valve, F, and controls the supply of steam to the pump drawing from the heater. When the water in the heater rises the float is raised, and the throttle valve opened, and, when the water level is lowered, the float follows, and the valve is closed; the level of the water is thus kept constant in the heater, and the pumps are completely filled with water. The regulating valve is a cock with a parallel key; the pressure of the steam keeps it perfectly steam-tight, although it may have worn slack in the shell; the pressure also keeps the shoulder of the key against the bottom of the stuffing-box, so that the stuffing gland is always kept slack. A relief valve and the necessary gauges are also fitted to the heater.

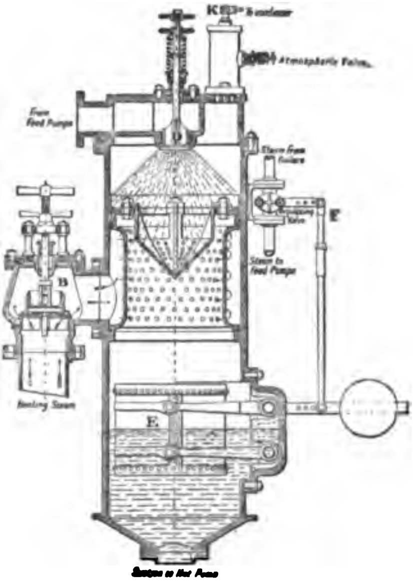

Fig. 195.—Weir's Feed-Water Heater and Automatic Regulating Gear.

Evaporators.—The advantages of supplying marine boilers with pure water are great, and are so obvious as not to need specifying. Samuel Hall, the inventor of the surface condenser, was so thorough in his desire to use only fresh water in the boilers of ships with his condenser that in 1833 he fitted evaporators to several of them made on precisely the same principles as govern the design of our modern ones. He, however, went beyond the modern engineer by placing his apparatus in the steam chest of the boiler so as to lose no heat. The necessity of it was not, however, so severely felt until voyages of considerable length had been made with ships whose boilers work at pressures of 100 lbs. and upwards. The weight of water evaporated in boilers, whose working pressure is 150 lbs., is much greater in proportion to the size than was the case with those working at 75 lbs.; and the evils arising from the deposit of scale are magnified with the higher pressure and consequent higher temperature. Again, the liability to put on scale is greater, inasmuch as the losses from leakages are greater with the higher pressures. Hence, the old system of making up loss of water by a supply from the sea, although a very simple and ready one, was not by any means satisfactory, and did not remedy the evil, but rather magnified it. The Admiralty and some private shipowners tried to obviate it by providing a

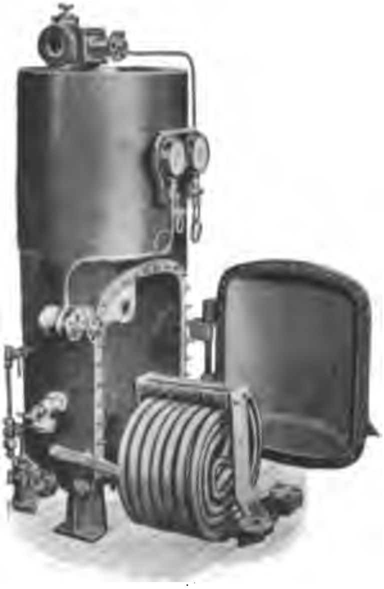

Fig. 196.

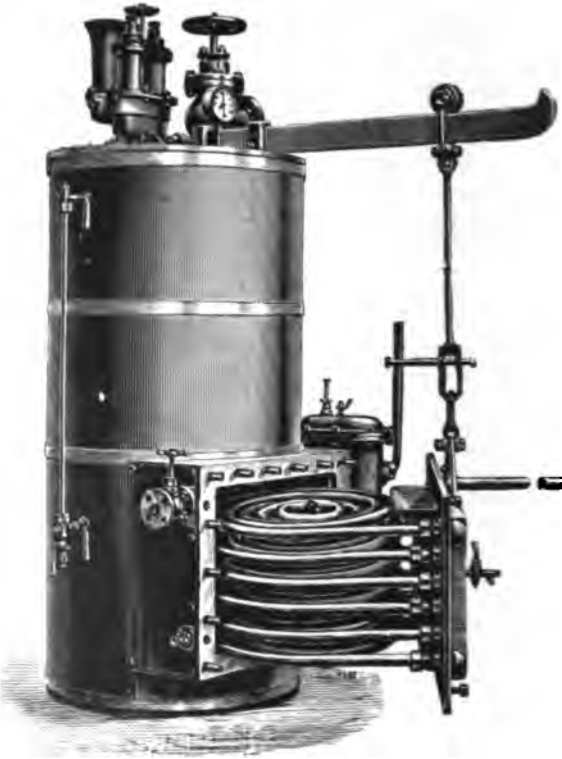

Fig. 196a.

supply of fresh water in the double bottoms, or in tanks specially fitted for the purpose. This, however, was only half a remedy, inasmuch as the fresh water generally obtainable contained large quantities of lime and other salts which gave a hard deposit more difficult to remove. Moreover, this fresh water cost money, and was so much extra weight to carry which also added to its cost. Then recourse was had to the auxiliary or donkey-boiler to obtain distilled water, which meant an expenditure of coal as well as labour in cleaning out these boilers after they had become coated. Besides, these small boilers very soon got so coated that they had to be stopped for a thorough clean-out; and during the time they were at work there was always the risk of damaging them. In spite of these difficulties, however, it was found to be the most satisfactory way of obtaining an extra supply for the main boilers, and, consequently, improvement was made in this direction by supplying a small boiler (Fig. 196), whose heat is obtained from either the steam direct from the main boilers or from the exhaust from one or other of the cylinders; the former plan was found eventually to be the best.

Fig. 196*a* may be taken as another example of the type now in general use. It consists of a vertical cylindrical shell fitted with mountings and gear similar to those on a steam launch boiler; instead of a furnace, combustion chamber, tubes, &c., it has a tubulous arrangement ingeniously contrived so that the steam is made to give up its heat to the water within the evaporator as far as possible, and the resultant water to drain away and be returned to the main condenser or hot-well. Steam is in this way raised in the evaporator, and passes from it to the main condenser or, as in naval ships, to the auxiliary condenser, the resultant water being finally pumped into the main boilers in the usual way. Sea water is pumped into the evaporator by a small donkey-pump, and the salt is blown down from the evaporator in the same way as was usual with boilers supplied with sea water. The internal tubulous apparatus is so arranged that it can be easily withdrawn from the shell, as shown in the figure, for a thorough clean-out when necessary. Instead of the steam from the evaporator being sent direct to the condenser, it can be made to do useful work by admitting it to the valve box of the low-pressure cylinder, as was done by Messrs. Weir originally.

There are other equally ingenious and efficient evaporators, but they are all worked on the same principle of heating water and converting it into steam with steam made in the main boilers. In all cases, as is only to be expected, tubes are employed in one form or another to effect this purpose.

Ladders.—The main ladder to an engine-room should not be less than 18 ins. wide, and where space permits should be quite 24 ins.; the sides are of flat iron bars usually, 4 × $\frac{3}{8}$ in.; the treads or steps are of cast iron, and 10 ins. apart, and from 4$\frac{1}{2}$ to 7$\frac{1}{2}$ ins. wide. The inclination of the ladder to the vertical is usually about 1 in 3 with narrow, and 1 in 2$\frac{1}{4}$ with broad steps; the hand rail is 1 in. diameter when of iron, and from 1$\frac{1}{4}$ to 1$\frac{3}{4}$ ins. when of brass; the former looks better from an engineer's point of view, is more durable, and easier kept clean. Ladders leading to the various parts of the engine are usually made lighter than the main ladder, and the steps are often formed of three "spills" or bars, $\frac{5}{8}$ in. diameter, or, better still, square section.

It is very essential that means be provided for the engineers to get easily and safely to every part of the engine requiring attention; these light ladders are a source of great convenience, and are amply paid for in the better attention given to the working parts.

Gratings and Platforms.—With the same object in view, good platforms to stand on, and gratings to form roads to the various parts should be provided. The engine-room platform is usually laid with either cast- or wrought-iron chequered plates, the pattern on whose face should be one which will give good foothold, and not prevent dirt from being swept from it. Some engineers prefer to have all bottom platforms made of wood, and laid over with sheet lead; the lead permits of good foothold, and is easily kept clean; water runs easily off it, and when in good repair it looks very well; but it is not so durable as the iron, and is very liable to damage from weights falling on it. The gangways leading to the upper parts of engines are sometimes made with chequered wrought-iron plates; but except when they are immediately over the working parts this is not a good practice, as both light and ventilation are obstructed by them, and they require constant cleaning. "Spill" gratings are, therefore, preferable in most cases, as they stop light and ventilation to only a very small extent, and require no cleaning; they are made with sides of flat bars $2\frac{3}{4} \times \frac{3}{8}$ in., and cross bars $\frac{5}{8}$ in. diameter, and spaced $2\frac{1}{2}$ ins. apart; those of large size and liable to support heavy weights have sides $3 \times \frac{7}{16}$ in., with spills $\frac{3}{4}$ in. diameter, pitched $2\frac{5}{8}$ ins.

Feed Filters are often fitted in ships of all sizes with the object of preventing any oil or grease getting into the boilers with the feed water; they certainly succeed in keeping out grease and solid matter, but fail to do so with oils. Generally the filtering material is Turkish towelling so placed in one or more layers that all the feed water must pass through it and leave deposited on it all solids and semi-solids, and to a great extent absorbed by it oils and greases in the liquid state. The apparatus is always arranged so that the "cartridge" of towelling can be easily withdrawn and a fresh clean one substituted without causing any stoppage of the engines.

Stokehole Ventilators should have an aggregate transverse area of 0·45 square inch for each pound of fuel burned per hour; or, say, 0·675 square inch per I.H.P. of trial trip in the Mercantile Marine, or 0·75 square inch per I.H.P. in Express steamers on short service and naval ships.

The area of the mouths should not be less than—

1·35	square inch	per pound	of fuel in	10	knot ships.
1·24	"	"	"	12½	"
1·13	"	"	"	15	"
1·03	"	"	"	17½	"
0·93	"	"	"	20	"
0·85	"	"	"	22½	"

CHAPTER XXIII.

WEIGHT AND OTHER PARTICULARS OF MACHINERY RELATING THERETO.

The Weight of the Engines and Boilers of every ship is a matter of extreme importance, although certain classes of shipowners rarely give a second thought to it, and probably would not pay anything extra for a saving of 5 per cent., although it would enable them to carry more cargo and gain yearly a respectable sum of money. In the case of small steamers of high speed it is of the very first importance; in fact such speeds as they are run at to-day are only attainable by the most careful designing of the engines, whereby the weight is reduced to a minimum. The best modern instances of this are to be found in the Torpedo Boat and Torpedo Boat Destroyer; some good ones are also observable in the cross-channel steamers, both paddle and twin screw. In the case of the modern Atlantic steamers the margin for weight of cargo is exceedingly small, so that a ship may have 20,000 tons displacement, and only a margin of 500 tons for cargo pure and simple. The machinery of such a ship will weigh 4,500 tons. It will be seen, therefore, that the saving of a small percentage of its weight would add a very large percentage to the deadweight cargo capacity. (Of course there is plenty of room for stowing cargo in these ships, so that they can carry plenty of goods, whose specific gravity is low, and thereby obtain high rates of freight if there is surplus displacement sufficient.) In the case of cross-channel steamers the present speeds were only possible by the reduction in size of boiler per I.H.P., consequent on the introduction of the triple and quadruple engine and forced draught. In their case, however, it was not merely a question of weight, but one of capacity as well. In the case of the cargo steamer the weight of the machinery is only a comparatively small fraction of the total displacement, and hence is, as already stated, too generally disregarded; but here, too, the question is worthy of attention, as in the times of bad freights the difference of a few tons in weight of cargo carried may make the difference between a profit and a loss.

Most modern engineers, however, have their attention called to the question of weight by other considerations than obtain with the shipowner.

In the case of warships, weight of machinery again is of prime importance, inasmuch as the armament and coal supply is in no small measure dependent on the margin of displacement left, after providing for the whole equipment and machinery. In other words, the lower the weight of the engines and boilers, the larger may be the armament, the armour, and the fuel supply. In the case of battle ships, whose speed is comparatively moderate, and the displacement on the dimensions, consequently larger than is the case with cruisers whose speed is very high, the machinery need scarcely be so light; and although, as a rule, the design of both engines and boilers is similar to that of the cruiser the rate of revolutions is not nearly so high.

Again, the small cruisers, whose speed is to be nearly as high as that of the big ones, have not the same margin for their machinery; it is, therefore, imperative that their engines should be lighter per H.P. The saving in weight is generally effected in this case by making the engines to run at a very high number of revolutions, with a high referred mean pressure, and designing the various parts with the greatest care possible, using only material of the greatest strength for the purpose. The factor of safety, therefore, is practically as high as that of other ships, because these ships may have to run in war-time continuously at or near full speed for long periods.

The machinery of the Torpedo Boat and Destroyer is, of course, also designed with the greatest care, and made of the very best material, but here the so-called margin of safety is reduced; in other words, engines are made so that at their ordinary rates of speed there will be as good a margin of safety as in any other ship; spurting they will have to do in time of war, but their coal supply does not permit of long duration of it, and the exigencies of the service in peace time would not require it to be done often in the course of a twelve-month; hence, although the factor of safety at high speeds may be low, it is still well within the safe-working limit of the material, and no great risk of accident is run. A reference to Tables xlix. to l*c*., will show the progress of modern engineering in both the Navy and mercantile marine during the past thirty years, and the lines on which modern engineers have advanced to attain their present success in coal consumption and the weight of machinery.

In 1870 Battleships and Cruisers were still fitted with simple horizontal engines and box boilers of low pressure. They were on the expansive principle, steam being cut off at a little past half stroke, and exhausted into a surface condenser. The boilers were designed for a pressure of 30 lbs., although sometimes 1 lb., or even 2 lbs. more was attained, and they bore generally a strong family likeness to fig. 130. The maximum piston speed was about 600 feet per minute on trial trips with large engines, and about 450 to 500 feet with the smaller ones. The mean pressure was about 20 lbs. per square inch in battle ships and 20 to 25 lbs. in cruisers. The I.H.P. per ton total weight machinery was about 7. 8·5 I.H.P. was developed for each square foot of grate, and 33 per 100 square feet of total heating surface; the consumption of coal was about 3½ lbs. at full speed, and about 2½ at half power.

To-day the battleship and first-class cruiser have triple-compound engines of the four cylinder type accurately balanced, so that vibration produced by them is very much less than that produced by the small early engines. These engines have a piston speed of nearly 1000 feet per minute, and revolve about 120 to 130 times in a minute. The steam is supplied to them at a pressure of from 200 to 250 lbs. per square inch, and the referred mean pressure is 44 to 50 lbs. Their I.H.P. is about 11 to 12 per ton of total weight of machinery at full speed, or from 9 to 9½ I.H.P. per ton at cruising speeds, corresponding to that of the ocean liner's full speed. The consumption of fuel is, however, not so greatly reduced in these ships as is the case in the mercantile marine, for although on trial trips it is at the rate of about 1·7 to 1·8 lbs. per I.H.P., on low cruising speeds it rises to as much as 2¼ lbs., including all auxiliaries, &c. The speed of battleships is 19 knots, first-class cruisers 24 knots, second-class cruisers 23 knots, third-class cruisers 21 to 22 knots. "Scouts," which are a cross between the third-class cruiser and the torpedo boat destroyer, are expected to attain a speed of 25 knots. The *Destroyer*, which commenced with a speed of 26 knots in 1894, attained 32 knots in 1900, and in the case of the "Cobra" and the "Viper," with Parson's turbine, 35 knots was reached. These very high speeds, due to the extremely light weight of both ships and engines, was obtained at a great sacrifice, consequently the British Government is at present content with a speed of 26 knots, and allow a weight of machinery 30 per cent. per I.H.P. greater than that with which the 30 knots were obtained.

Piston Speeds.—In 1870, prior to the introduction of the compound engine, the piston speed of an Atlantic passenger steamer was 400 feet per minute, of smaller passenger steamers 350 feet, while fast cruisers had a piston speed of 550 feet; battleships the same, while small cruisers had only 450 feet.

In 1880, before the advent of triple-compound engines, Atlantic and ocean steamers generally had advanced to 600 feet, the smaller passenger steamer to 450 feet, the cruiser to 600 feet, the battleship to 600 feet, and small cruisers to 550 feet per minute.

In 1882 forced draught was introduced into the Navy, and by this means the increased supply of steam permitted of higher speeds, so that in 1885 large cruisers had a piston speed as high as 660 feet, and small ones as high as 750 feet per minute. At that time the triple engine had got a foothold in the mercantile marine, and permitted of a piston speed of 750 feet in ocean mail steamers and small passenger steamers 550 feet per minute.

By 1890 higher pressures of steam had been employed, and confidence in the triple engine established; consequently the piston speed for cruisers and battleships was raised to 800 feet, ocean liners to the same speed, and small passenger steamers in some few instances exceeded this.

Five years later battleships had a piston speed of 875 feet, cruisers a little more; while the newcomer, *Destroyer*, attained its very high speed by moving the pistons at a velocity of 1200 feet per minute. The Atlantic mail and the small passenger steamer had made only very slight advances on 1890.

To-day we find the battleship and cruisers with four crank carefully balanced engines running at 960 feet, and small cruisers at 1000 feet per minute; the Atlantic mail steamer at 940, and the small express steamer at 950 feet. The cargo steamer pure and simple makes her voyages with a piston speed of 500 to 660, depending on the size of the engine.

The increase in piston speed has been obtained by quickening the revolutions from 70 to 120 in battleships, and from 75 to 140 in large cruisers; the greatest difference is, however, in small cruisers, the change being from 90 to 250 revolutions. In the mercantile marine the revolutions of an ocean liner is 80 against 55, and in ordinary small passenger steamers 150 against 80 to 90 of twenty-five years ago. It is therefore not surprising to find that the I.H.P. per ton of machinery in ocean mail steamers is only 6·75 as against 5 or 35 per cent. increase, while in large cruisers it is 12 against 7, or an increase of 70 per cent. In small cruisers the I.H.P. is now 19 per ton against 6 of twenty-five years ago, or an increase of 216 per cent. It may be taken as a rule that a reduction in weight per I.H.P. is obtained by increased piston speed if it is due to high revolutions; on the other hand, if piston speed is got by lengthening the stroke, no saving is effected, but, on the contrary, an increase may be looked for.

Boiler and Mean Pressures.—The general tendency of increase in boiler pressure is to increase the weight of the engines as well as the boilers. If, however, an increase of mean pressure is obtained by the increase of boiler pressure, and the rate of increase be greater than that of the boiler pressure, the weight of the engines per I.H.P. may be really decreased. Such, however, was not generally the case in the history of marine engineering progress. In 1870 with a boiler pressure of 45 lbs. absolute, the mean pressure was as much as 24 lbs. in the expansive surface condensing engines; or 53 per cent. of the boiler pressure. In the compound engine of about that time the referred mean pressures was only about 22 lbs. with the boiler pressure 75 lbs. absolute, or only 29 per cent. Later, with a boiler pressure of 105 lbs. absolute, the mean in express steamers was only 30 lbs., or 28·5 per cent. Later still, with the triple engine and a boiler pressure of 170 lbs. absolute, the mean in cruisers of high speed was 38 lbs. or 22·4, and 32 or only 18·8 per cent. in merchant ships. To-day, with initial pressures of 250 lbs. in the Navy and 215 lbs. in the mercantile marine, the referred mean pressure is under 20 per cent.

The effect of increase of pressures is best seen by comparing the second class cruiser of 1898 with that of 1896. These engines developed practically the same power at the same number of revolutions; the design of each was identical, but the cylinders were made smaller in diameter for the increase in mean pressure, the referred mean pressure being 50·9 lbs. in one case and 40·3 in the other. As might be supposed, the weight of engines is practically the same in each case. If, however, the boiler pressure had been increased without an increase of referred mean pressure, the I.H.P. would have been the same, but the weight of engines greater. If the boiler pressure had remained unaltered but the referred mean pressure increased, there would be an increase in I.H.P. without a corresponding increase in weight. It may therefore be assumed that increases in boiler pressure only add to the weight of the *engines* when there is no substantial increase in the referred mean pressure. Also, that when there is a substantial increase in the referred mean pressure, the weight of *machinery* per I.H.P. is not necessarily increased by the increase in the boiler pressure.

Materials.—The weight of machinery is, of course, very considerably influenced by the materials of which it is made. For example, the ordinary mercantile engine is largely made of cast iron of a quality whose strength does not exceed 7 tons per square inch. The engines of naval high-speed ships have little cast iron in their construction, but what there is can withstand 11 tons per square inch, and some of it even as much as 13 tons; much of them is of cast or wrought steel, having a strength of 28 tons in place of the merchants ships' 7-ton cast iron. Again, in naval ships the bronzes are extensively used in lieu of cast iron, and although Admiralty bronze is but little stronger than best cast iron, it is much more so than the 7-ton castings, and consequently may be, and is from one-half to two-thirds the thickness of the corresponding iron castings of the mercantile marine. The light weight per I.H.P. of the cross-channel mail and passenger steamer is largely due to their having engines made of the same materials as naval engines. New materials are from month to month brought to the notice of engineers by metallurgists which promise great things in the way of saving weight, and some of them fulfil the promise, but others, from one cause or another, disappoint both their discoverers and users. Aluminium is an instance of this, although in time it may redeem its character and enable the engineers to carry out all the good intentions they had when the price fell to 2s. 6d. per lb. At present all hopes are centred on nickel as the improver not only of steel, but of aluminium. Nickel steel is serving the turn of the naval designer by withstanding the impact of shot; may it not serve the turn of the engineer by withstanding high stresses—especially alternating ones—and any number of repetitions of such shocks as come on many of the moving parts of the marine engine?

Annealing and oil tempering, too, may be more extensively used to improve steels as now made, or at anyrate to make them more uniform in nature and with more of the valuable character as usually described by its friends than as found in practice.

The superior bronzes, too, may be used with advantage to save weight, inasmuch as their ultimate strength is quite equal to that of steel; moreover, castings may be made of them much thinner than if of steel. Most of these bronzes can be forged and rolled, when their strength is improved, and they are rendered even more suitable for high-class engineering than in the cast state.

Design has great influence on the weight of marine machinery, as may have been supposed; and what can be done by the most careful designing is seen in the case of the machinery of the *Destroyer*, when 45 I.H.P. per ton is developed, or even in that of the third-class cruisers, when 20 I.H.P. are developed under forced draught, and even 14·5 under natural draught per ton weight.

In the case of the *Destroyer* it will, of course, be urged that the saving was effected largely at the expense of strength. This cannot be said of the cruiser's engines any more than it can be of those of the cross-channel steamer.

It is true that not much saving can be effected in the principal working parts, except by such means as boring the shafts and rods. No doubt now, when improved tool steel and appliances admit of boring at a cheap rate, hollow shafts will be used as freely in express steamers as in naval ones, and with the same satisfaction. The great saving will be effected by change in design of the fixed portions, and in most of the smaller details being dealt with drastically. It astonished many marine designers who turned from large engines to the quick ones in *Destroyers* to find how much material was practically wasted in the numerous unconsidered fittings and fixings of an engine and engine-room.

Not much in any case can be done in reducing the weight of the parts subject to *alternating* stresses, except by the use of materials better calculated to withstand them, such as nickel steel, high carbon steel, &c., oil tempering, &c. Some saving, however, can be made on those parts liable only to *intermittent* stresses, especially when of a definite and easily calculable nature; and still more may be done in this direction on the parts subject to simple, steady loads.

Table xlviii. gives the safe limit of the various metals used by engineers under the several conditions obtaining in practice, and the following observations may be taken as guides in dealing with the question of design:—

Effect of various Loads and Stresses.—The various parts of marine machinery are subject to loads of different natures, and each one, as a rule, has to bear a repetition of its own particular load constantly and continuously. The nature of the load should determine the magnitude of the greatest stress which the part subject to it should sustain. If it be a steady one all materials will withstand much higher stresses than are possible with a load more or less suddenly applied and removed continuously. Experience has shown that in structures on land metal can be loaded so that the stress is considerably more than half its elastic limit, if it is steady—that is, if there is no variation; on the other hand, it has been found that if a load of such magnitude is applied and removed quickly and continuously the metal soon gives way. If the stress is lowered either by reducing the load or increasing the section of material to sustain it, the number of applications of load to produce fracture is larger; if further lowered, it takes a larger number still; finally, it is found that if the load does not stress the material beyond a certain fraction of the elastic limit, the number of applications may be so large without producing fracture that the load may be considered quite a safe one, and the corresponding stress used for calculations when employing that material under similar conditions. Such loads as these may, for convenience, be called *intermittent*, and the stresses produced *intermittent stresses*.

The parts of the engine subject to intermittent stresses are the piston-rod ends in and beyond the pistons and cross-heads, the piston-rod and connecting-rod caps and bolts, the cylinder cover studs, main bearing caps and bolts, the bolts and studs of the valve gear, the guides, &c., the crank-shaft, the high-pressure cylinder and its covers, the medium-pressure cylinder and its covers generally, and all the column and framing bolts. The tunnel or intermediate shafting theoretically has a *steady load*, but really it is moderately intermittent—that is to say, the variation is not from zero to the maximum, but from a minimum considerably above zero to the maximum.

If the load varies from a positive to a negative value—that is, from tension to compression and from compression to tension—the load is said to be an *alternating* one and the stresses on the material *alternating stresses*. Wohler, of Berlin, has shown, and Professor Unwin, in this country, supports the theory, that the effect of a positive and negative load acting alternately continuously is nearly the same in its destructive effect as a positive load of the same magnitude as the two jointly. That is, if a piston-rod is subject to a thrust of 20 tons and a pull of 15 tons or *vice versâ*, the effect on the material is nearly the same as if it were subject to an intermittent load of 35 tons. It is, however, not quite the same, and further experiment with modern metals may show that at moderate loads the difference between alternating and intermittent is not so great. In any case, material subject to alternating loads must not be stressed so highly for any one direction as if those loads were intermittent; in fact, from Wohler's experiments the stress should not be much more than half. The rapidity of intermission and alternation is also a factor in determining safe stresses for materials.

Further, the shape of the part subject to load is of importance; if there are incisions similar to that of a screw thread, which suddenly change and concentrate stress and invite fractures the stresses should be much lower than those of a plain uniform section, such as that of a bar could safely bear.

The parts of an engine subject to alternating stresses are the piston-rods, connecting-rods, cross-heads, columns, pistons, low-pressure cylinder, its cover and bottom, the medium-pressure cylinder, &c., when at low powers, paddle shafts and screw shafts at their outer ends due to the weight of propellers and the reaction of floats, or blades, especially in the case of a screw ship when racing or running light. The valve motions, the air-pump levers, rods, &c., are subject to partially alternating stresses.

Table xlviii. gives the safe limits to which an engineer may go under the various conditions, if weight is of prime importance in designing engines for continuous work. If weight is not an object of first importance a reduction of 10 per cent. may with advantage be made. If the article is nicked, suddenly changed in section, or has a screw thread, a reduction of 25 per cent. should always be allowed. In the case of alternating stresses the equivalent force one way only is taken, for calculation—viz., tension.

TABLE XLVIII.—Limits of Safe Working Stresses on Metals (Tension).

Metal or Alloy.	Nature of Load.	At Revolutions per Minute, not exceeding					
		60	100	200	300	400	500
Iron castings, good quality, - - -	Steady, -	5,000	4,920	4,800	4,700	4,600	4,500
	Intermittent,	4,500	4,420	4,300	4,150	3,880	3,600
	Alternating,	3,300	3,280	3,220	3,110	2,920	2,700
Iron bars and plates, good, - - -	Steady, -	14,000	13,850	13,800	13,100	12,600	12,000
	Intermittent,	12,600	12,410	11,920	11,350	10,650	10,000
	Alternating,	9,850	9,720	9,400	8,950	8,500	7,950
Steel bars and plates, mild, - - -	Steady, -	16,500	16,400	16,100	15,720	15,320	14,800
	Intermittent,	14,850	14,720	14,200	13,500	12,750	11,900
	Alternating,	10,900	10,800	10,450	10,000	9,500	8,900
Iron and mild steel forgings, - -	Steady, -	15,000	14,900	14,660	14,300	13,900	13,500
	Intermittent,	13,500	13,400	12,930	12,300	11,600	10,800
	Alternating,	10,000	9,910	9,600	9,200	8,700	8,200
Steel castings, good ordinary, - -	Steady, -	12,000	11,950	11,750	11,530	11,200	10,800
	Intermittent,	10,500	10,400	10,380	9,600	9,060	8,500
	Alternating,	8,000	7,920	7,700	7,350	6,950	6,500
Steel high carbon (·45 per cent.) forgings, oil tempered, - -	Intermittent,	17,000	16,730	16,210	15,450	14,550	13,600
	Alternating,	12,500	12,400	12,000	11,550	11,000	10,300
Nickel steel (3½ per cent.) forgings, -	Intermittent,	24,000	23,750	22,950	21,850	20,550	19,000
	Alternating,	17,800	17,650	17,100	16,400	15,500	14,600
Nickel steel, mild, plates and bars, -	Intermittent,	19,400	19,220	18,550	17,700	16,650	15,500
	Alternating,	14,200	14,100	13,650	13,050	12,350	11,600
Nickel steel, high tensile, plates and bars, oil tempered, - -	Intermittent,	28,000	27,750	26,850	25,600	24,100	22,400
	Alternating,	21,000	20,800	20,200	19,400	18,400	17,250
Bronze, Admiralty mixture, castings, -	Steady, -	7,650	7,600	7,500	7,350	7,100	6,800
	Intermittent,	6,800	6,750	6,500	6,210	5,880	5,450
	Alternating,	5,000	4,970	4,850	4,540	4,360	4,000
Phosphor bronze castings (wheel brand),	Steady, -	10,000	9,840	9,600	9,400	9,200	9,000
	Intermittent,	9,000	8,840	8,600	8,300	7,960	7,500
	Alternating,	6,750	6,650	6,500	6,300	5,900	5,400
Phosphor bronze castings, forged and rolled, - - -	Intermittent,	20,000	19,720	19,200	18,300	17,280	16,000
	Alternating,	15,000	14,880	14,500	13,930	13,300	12,600
Naval brass, rolled (hard), - - -	Intermittent,	16,750	16,650	16,000	15,200	14,300	13,400
	Alternating,	13,800	13,660	13,160	12,550	11,850	11,100
Do., do., annealed,	Alternating,	9,000	8,840	8,600	8,300	7,760	7,500
Stone's manganese & aluminium bronze, Delta, Bull's & Aich's metals, cast, - -	Steady, -	12,500	12,450	12,250	12,000	11,700	11,250
	Intermittent,	11,000	10,850	10,430	9,960	9,400	8,800
	Alternating,	8,000	7,920	7,700	7,350	6,950	6,500
Do., do., wrought,	Intermittent,	20,000	19,720	19,200	18,300	17,280	16,000
	Alternating,	15,000	14,880	14,500	13,930	13,300	12,600
Do., do., annealed,	Intermittent,	14,850	14,720	14,200	13,500	12,750	11,900
	Alternating,	11,000	10,860	10,430	9,960	9,400	8,800

Shearing and Torsion.—Certain metals have a very high power of resistance to tension and compression without the like power to resist shearing; for example, naval brass has a coefficient of shear of only 55·8 per cent.—that is, its strength against shearing is little more than half what it possesses against tension; on the other hand, cast iron has a coefficient of 162 per cent., and phosphor bronze castings 113 per cent. The coefficient of wrought steel and iron is 80, while that of most of the strong zinc bronzes is, as a rule, only about that of naval brass. They are consequently not very fit for such purposes as shafting, where shearing forces are always of importance, especially so in the case of tunnel shafting.

The Elastic Limit or Yield Point.—It is probable that every material which has never been subject to direct tension will have some degree of permanent set after application of any load; it will, of course, be very small, and generally microscopic. If, however, the material is stressed nearly to what we call the elastic limit, released, and again subjected to stress, there should be no further permanent set till that point (the elastic limit) is reached.

When this limit is found there is reason to suppose that so long as *no portion* of the material is subject to a stress in excess of it, the material should endure for ever. Unfortunately materials do give way from what is called "fatigue" without apparently having been subject at any time to a stress nearly approaching the supposed elastic limit. It follows, then, that the safe limit of stress is considerably below the elastic limit, or that the loads on *some portion* of such structures as have collapsed, are much greater than generally supposed. It is highly probable that the latter is the case, and to a great extent the cause of accident. In a complex structure it is nearly impossible to exactly proportion each part to its load, and absolutely impossible to spread the sustension of load evenly over every inch of area of section. Moreover, in the process of manufacture initial stresses are often set up in many places, all of which are of uncertain magnitude and direction; to these are added the stresses produced by working load, which may not be, and probably are not, distributed in accordance with theory. It is, therefore, far from improbable that a structure designed with a theoretical margin or factor of safety of ample magnitude has not in one or more places material stressed dangerously near to the elastic limit, perhaps sometimes even beyond it.

Safe Working Stresses.—The limits given in Table xlviii. may be used when the maximum stresses are calculable and distributed uniformly over the sections. For example, a weight suspended by a bar having an eye at each end, so as to hang freely, will cause a uniform stress on the transverse sections; if instead of eyes forged with the rod the rod was screwed and fitted with nuts whose faces are rough and the surfaces on which the nuts sit are uneven, the stress will not be uniform, and may be much greater on one side than the other. Again, the sections of a round or square bar loaded and subject to bending either as a beam or cantilever are not uniformly stressed, the outer layer, or that furthest from the neutral axis, being much more so than those nearer it; but at the same time, the layer being firmly attached to its neighbour, receives support from it in a different way from what is the case of a fagot of strips free to slip one on the other as in a carriage spring. The rules for estimating are based, however, practically on the latter supposition, hence what would seem to be a too high stress can be safely borne by such parts.

In the case of a complex article like a crank-shaft subject to complicated loads, it is impossible to estimate the maximum intensity of stress at any and every part of a section. It is certain that at very few, if any, places the stress is uniform over the whole section at such places; and at some the intensity of stress must be great on portions of the sections. Moreover, the quality of the material is not uniform throughout such a forging, the strength across the direction of flow of metal being less than given in Table xlviii. Under such circumstances the nominal stress adopted for calculations should be much less than those given in that table. For similar parts of light machinery the nominal stress may be taken at 60 per cent. of those given in Table xlviii., and for those of ordinary machinery 35 to 40 per cent. In practice, therefore, the designer must carefully consider each case by itself, with all its conditions and surroundings, to determine the nominal stress on which to calculate the dimensions of the parts which must sustain the load.

Stretch under Tension.—Rigidity of structure is often of prime importance, and generally so in the engine-framing, guides, &c. All materials stretch more or less under stress, and the lower the stress the less is the stretch; consequently it is often found necessary to make details of the structure larger than necessary for mere strength, in order to avoid the stretching of them, which would be inconsistent with rigidity or stiffness of form. It is generally forgotten that up to 7 or 8 tons tension per square inch cast iron stretches more than steel or wrought iron, and that the material has practically no elastic limit, for it has no permanent set up to fracture.

Again, there are other conditions which must not be overlooked, as, for instance, in the case of castings as distinguished from forgings or other wrought forms of material. The latter are practically free from internal defects when machined and made part of an engine; the castings, however, *may* be solid, but are often not so, and must not be assumed to be. To allow for such contingencies a reduction of 10 to 20 per cent. should be made from the stress given by sound bars in the testing machine as the elastic limit of castings. The figures in Table xlvii. for casting are based on this allowance. Again, rough forgings often have incipient flaws arising from the indenting of the hammer in forging, and, in the case of steel, from small gas bubbles and scoria near the surface of the ingot, and sometimes from slag getting forged on the metal. In the bright working parts such imperfections are machined away.

Another serious consideration in determining the nominal stress for calculation is the purpose the engine has to serve. Express steamers, such as passenger and mail boats, have to exert 80 to 90 per cent. of the trial or maximum power for 50 to 70 per cent. of the year; cross-channel steamers (short runs) steam about 25 to 35 per cent. of the year; while excursion and river steamers do from 30 to 50 per cent. They all have occasionally, and are always liable to have, to run at full power. Every part of the machinery of such ships should, therefore, be much lower stressed at full power than the corresponding parts of machinery which is seldom or never worked to the full power for which it was designed, whose full speed on service is done with only 75 per cent. of such power, and that for only 10 per cent. of each year.

The speed of the engine also affects the problem. A slow-working paddle-engine, with few reversals of motion and time for "cushioning" and other means of avoiding shock, does not jar its component parts as does a high-speed one, the momentum of whose moving parts materially influences the stresses on them and their connections, which stresses are most suddenly applied and removed or reversed many more times in a minute, and which may be increased by water in the cylinder to an unknown extent, and from want of time and other causes may not get the cushioning at the right periods to do good and ease the concussions. Such engines, however, are generally required to run at full speed for very short spells and those far apart, and may, therefore, have lighter scantlings than would be necessary for a similar engine on shore, such as an electric light or power engine, which has to run every day for five to nine hours, or even more.

Paddle Engines, when required to be very light, may be designed by referring to the stresses given under 60 revolutions and modifying on the lines laid down above. When weight is not of so much importance as endurance, the stresses may be reduced by 10 per cent. all round.

Naval Engines, Express Steamer Engines, and **Ordinary Cargo Steamer Engines** may be designed by taking the figures under 200 revolutions and allowing 10 per cent. reduction all round, and modifying on the lines laid down.

For **High-speed Craft**, such as **Torpedo Gunboats, Destroyers, Third-class Cruisers, &c.**, the column under 300 revolutions with the modifications, but without the all-round 10 per cent. reduction, may be followed.

For **Tramp Steamers**, whose power is small, so that the engines are virtually always working at full speed, and made of materials untested and generally of common quality, a reduction of 20 per cent. from column 100 or 200 revolutions is advisable when calculating the various parts of the engines and machinery.

Boilers, &c., form the most important part of the machinery from the weight point of view, and the possible saving is greater than from any other part, as has been seen when water tube boilers have been substituted for cylindrical, and earlier when steel shell plates took the place of iron ones. The Admiralty have reduced the thickness of these plates till little remains possible in that direction with ordinary steel, but it is to be hoped that the Board of Trade, Lloyd's, and other authorities will shortly follow suit. It is also equally desirable that a material as superior to the present mild steel as it was to iron will soon be placed on the market for the use of boilermakers. It is possible, however, to make a much lighter boiler without running any risk beyond what already exists. For example, the steel bars for stays are, when the boilers are in use, subject to a steady continuous load, gently applied and released; they are tested mechanically under similar conditions to a stress of 30 tons per square inch; but when at work are stressed only to 4 tons, whereas 6 tons would not be excessive, especially in these days of non-corrosion in the inside of boilers. With a reduction in the size of stays a saving would be made on the nuts, &c. The inner plating is also heavier than it need be under the above conditions. The heads and snaps of rivets are made regardless of weight, and the fact that most of the rivets are subject to shear only. For cleaning purposes boilers were and are still designed with room in various parts sufficient to admit a man and permit him to do the work. This means a larger boiler and much more water than is necessary. Now, with salt water cut off and practically pure water used, there cannot be the risk of scaling, especially in short voyages, to warrant such extravagance in space. Locomotives use hard water and water containing much solid matter, but no such provision is made as in ordinary marine boilers for excavating scale. Of course where there is a strong probability of salt water being used some such provision must be made, but this would be only in ships where weight of machinery is unimportant and long voyages the rule.

Cost.—Although less material is used when the weight of an engine is cut down, the value of that saved is far exceeded, as a rule, by the cost of labour employed in effecting the reduction. The substitution of one material for another with the object of saving weight invariably means a more costly one. The removal of surplus materials from all forgings, and some of that even from castings, is a costly process, although the modern machine tool, with the high-class self-hardening steels now in use, permit of it being done at a much cheaper rate than formerly obtained. Also, it is generally found that the reducing to a minimum of the sizes of flanges, facings, &c., entails much more care in moulding, marking off and machining, with the consequent greater cost of labour; besides which, more costly methods of doing work are necessitated by the want of room in carrying it out. Then, too, the much higher number of revolutions resorted to with the object of reducing weight per I.H.P. requires much greater care in the manufacture and adjustment of all the working parts, and the use of fittings and appliances which were done without before.

A compound naval engine twenty-five years ago cost about £12 per I.H.P., or £75 per ton. Ten years later triple engines of large size were £6·4 and small £8 per I.H.P., and still about £75 per ton. In 1890, the cost of cruisers' machinery was about £7 per I.H.P. and £90 per ton. Latterly large cruiser machinery with Belleville boilers costs about £12 per I.H.P. and £130 per ton.

The very light naval engines of the past few years illustrate what can be done in the way of saving weight when cost is of secondary consideration. At the same time it will be seen that the cost per I.H.P. on trial at full power is really low, and even if the lower power or that developed without forcing be taken, it is seen that these engines by comparison are by no means costly. For example, the third-class cruiser engines of 1891, with cylindrical boilers, cost £6·75 per I.H.P. full power, at natural draught it was only £10, or £93 per ton. The third-class cruiser of 1897 has water-tube boilers and engines costing £6·5 per I.H.P. of full power, or £9 of natural draught; this was, however, £118 per ton. The destroyer engines of 1895 cost only £6 per I.H.P., but £200 per ton; later ones cost as much as £236 per ton.

Taking similar periods for the mercantile marine, an ordinary compound engine twenty-five years ago cost about £7·5 per I.H.P., or £35 per ton. Ten years later the triple engine was costing £6·5 per I.H.P., or £36 per ton. In 1890, when prices were high in consequence of the Naval Defence Act having filled most of the shipyards and engine works with orders, engines could be got for under £6 per I.H.P., but still £36 per ton. Ten years later the engines of cross-channel steamers, whose weight would be only about 220 lbs. per I.H.P., could be purchased at £5·5 per I.H.P., or £53 per ton, and heavy engines at about £6·5 per I.H.P., and £35 per ton.

Comparing the cross-channel boats' machinery with that of the 1891 third-class cruiser, it will be seen that per I.H.P. they cost the same money, but the cruiser engines weigh only 156 lbs. against the 220 of the merchant ship. The latter costs therefore only £53 per ton against the £93 of the Naval ship.

Auxiliaries and Appurtenances.—Twenty-five years ago the marine engine was practically self-contained; it had as a parasite an auxiliary feed donkey pump, and in large ships two such pumps. The general use of water ballast later on necessitated another and a larger donkey pump, and feed heaters were found to be good things; centrifugal circulating pumps also came to be generally used in large ships about the same time. In the Navy, in consequence of the high revolutions, it was found better to have independent feed pumps, and, later, to take from the main engine all pumps except the air pump. This practice was soon followed in the mercantile marine with fast running engines. Then fresh water for making up the water from the boilers was deemed necessary, and evaporators or distillers were used. Starting and turning engines were known then, but have been used very generally since; they are now regular fittings in ships of all sizes. Feed-water filters are common where ships are engaged in long voyages and use oil in the cylinders. All these things add to the weight of the machinery without contributing anything directly to the I.H.P. used as divisor. The I.H.P. developed in the auxiliary machinery of a warship is now enormous, and can be gauged by referring to Table xxxiv. In a large merchant steamer, especially long voyage express ones, the auxiliaries add very much to the weight and steam consumption.

Spare Gear also adds to the weight, and is more in extent to-day than was formerly the case in the mercantile marine; but less in the Navy, where in old days nearly half an engine was carried (*v.* Appendix C).

Fuel.—The consumption of fuel is another most important factor in all considerations of weight; especially is this the case where, as in the North Atlantic, the distance between coaling stations is long. It is no use paring down the weight of the machinery if by so doing the fuel consumed per I.H.P. is increased. The great cry of the opponents of the water-tube boiler is that the increased consumption of fuel outbalances the saving in weight, so that, for a given voyage, the total weight of machinery and fuel together is greater with a water-tube boiler installation than with a cylindrical one with Howden's or Ellis' forced draught. Twenty-five years ago 2·5 lbs. per I.H.P. was the general rate of consumption of good coal; a little later it was as low as 2 lbs. with the very large three-crank compound engines. The triple engine in its early days consumed 1·7 in a general way, and 1·5 under best conditions. Now the best engines with forced draught and hot air, feed heaters, &c., consume 1·25 lbs., and ordinary good triples and quadruples 1·4 to 1·6 lbs. of good mercantile coal.

In the Navy the consumption of coal (including auxiliaries) was with triple-compound engines and the cylindrical boiler 1·8 lbs. of Welsh coal when running full speed. To-day it is from 1·8 to 2·2 lbs. with the water-tube boilers under sea conditions. If the I.H.P. of the auxiliaries were included the rate would, of course, be somewhat lower in both cases.

Relation of Weight to Tonnage.—Taking the same period as before, it is interesting to note that in Atlantic mail steamers the machinery was 13 per cent. of the displacement, and the I.H.P. at the rate of 1·1 per ton of gross register. In small passenger steamers it was about 13 per cent. and 1·35 per gross register ton.

Ten years later the speed of ocean steamers had increased by 20 per cent. The weight of the machinery was still about 13 per cent. of the displacement, but the I.H.P. was 1·8 per gross register ton.

The increase of speed with the smaller steamers was 25 per cent., the weight of machinery 15 to 19 per cent. of the displacement, and the I.H.P. 1·5 to 1·7 per gross register ton.

In 1895 the Atlantic express steamer had a speed of 21 knots obtained with machinery whose weight was 20 per cent. of the displacement, and whose I.H.P. was at the rate of 1·9 per gross register ton. Twin screw cargo and passenger boats with a speed of 15 knots of this period had a weight of machinery of only 7·3 per cent. of displacement and less than 1 I.H.P. per gross register ton, while in similar boats of 13 knots speed the proportions were 5 per cent. and 0·5 I.H.P.

Fast Channel screw steamers of 17½ knots were at the same time on service with machinery weighing 27 per cent. of the displacement, and indicating 3 to 5 H.P. per gross register ton.

To-day 23·5 knots is attained by the fastest express Atlantic steamers by means of machinery requiring 23 per cent. of the displacement to carry it, and developing 2·3 I.H.P. per gross register ton.

A speed of 25 knots is now aimed at by the Cunard Company to compete with the magnificent German steamers at present taking the lead in the Atlantic.

In the Navy a quarter of a century ago a speed of 18·5 knots was attained on trial trip by machinery whose weight was 27 per cent. of the first-class cruiser's displacement. Ten years later the same speed was attained with triple-compound engines only 15·5 per cent. of the displacement in weight. To-day 17·7 per cent. is allotted for a speed of 24 knots; in second-class cruisers 15 per cent., and in third-class cruisers 16·8 per cent. for speeds of 20 to 21 knots. The destroyers' machinery for 30 knots has a weight equal to 46 per cent. of the displacement.

The Express Ocean Steamer to-day is of enormous size, even larger than the "Great Eastern," which, until quite recent times, was deemed to have been a huge mistake on account of her size. The speed of the "Great Eastern" was, however, very small compared with that obtained by the magnificent steamers recently placed on the Atlantic service by German enterprise, and which attain at sea an average speed of over 23 knots with a consumption of coal not exceeding 1·3 lbs. per I.H.P. By referring to fig. 11, it will be seen that the designers of the machinery of the "Kaiser Wilhelm II." have not been unmindful of the weight that was to be saved by careful design of the fixed parts, such as the columns and foundations, and it is no doubt due to this that these engines can develop 7 H.P. per ton of total weight of machinery in spite of the high boiler pressure of 220 lbs. To similar causes may be traced the fact that the French designers of "La Savoie" could obtain 9 H.P. per ton.

The Cape, Australian, and the East Service is now generally performed by steamers of somewhat smaller size than those on the Atlantic, and whose trial trip speed is about 18 knots.

In Cross-channel Express Services we find some of the most interesting examples of Naval architecture and marine engineering, most of which show that by following on the lines of Naval design, but not going to its full extent, very high speeds are safely and satisfactorily obtained. Messrs. Laird Bros. have shown that even a speed of 23½ knots can be maintained on service with such machinery in a ship of only 2700 tons displacement. To-day this class of steamer is to be found on the North Sea, English Channel, and Irish Channel, doing daily and nightly service at speeds varying from 17 to 20 knots, with displacements ranging from 1250 to

2500 tons, by having engines indicating 9 to 10 H.P. per ton of total weight of machinery, and whose fuel consumption is only $1\frac{1}{2}$ to $1\frac{3}{4}$ lbs. per I.H.P.

For Shallow Water Ports and River Service the paddle engine has continued to hold its own (although twin screws could be used in most instances) by their constructors so designing the engines that 8 to 9 H.P. is now developed per ton of total weight, as against the 6 to 7 H.P. of the light paddle engines of thirty years ago, with a fuel economy that will compare favourably with that of screw engines.

The following table gives the conditions now ruling which affect the question of weight, as well as the weight of the various engines now in use on shipboard :—

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TABLE XLIX.—PRESENT CONDITIONS OBTAINING WITH MARINE MACHINERY.

	Express Steamers.			Steamers for Cargo and Passengers.			Cargo Steamers.			Naval Ships.				Paddle Steamers.		
	Atlantic.	General Ocean.	Cross Channel.	Atlantic.	Medium.	Small.	Large.	Medium.	Small.	1st Class.	2nd Class.	3rd Class.	De-stroyers.	Large.	Medium	Small.
Displacement, tons,	30,000	15,000	3,000	30,000	10,000	3,000	30,000	10,000	5,000	15,000	6,000	3,000	360	5,000	1,000	500
Speed on trial, - -	24·0	19·0	21·0	17·0	15·0	13·0	12·0	11·0	10·0	24·0	21·0	20·5	30	22·0	19·0	16·0
Revolutions, - -	80	80	175	85	80	100	70	80	90	140	180	250	390	50·0	52·0	55·0
Piston speed, - feet,	950	800	900	800	700	700	600	600	550	980	900	1,000	1,170	700	600	400
Boiler pressure, - lbs.,	220	200	180	200	200	180	180	170	170	205	300	300	250	140	120	120
Referred mean pressure ,,	36·0	38·0	45·0	39	36	34	33	30	30	45	50	52	47	42·0	40·0	38·0
Referred mean ÷ absolute boiler pressure,	·153	·177	·236	·166	·167	·174	·169	·162	·162	·205	·156	·165	·177	0·300	0·333	0·316
Total weight of machinery per I.H.P., lbs.,	336	330	224	490	438	336	408	400	336	215	190	120	51	300	280	300
Weight of machinery ÷ displacement,	0·23	0·125	0·268	0·096	0·060	0·125	0·042	0·050	0·047	0·200	0·148	0·178	0·460	0·360	0·375	0·400
I.H.P. per ton of machinery weight,	6·75	6·80	10·0	4·6	4·880	6·75	5·50	5·60	6·67	8·36	9·36	15·2	...	...	...	...
I.H.P. per sq. ft. of grate,	13·0	11·50	13·5	12·5	12·5	12·5	11·75	11·50	10·5	10·72	9·84	16·24	...	...	...	...
I.H.P. per 100 sq. ft. of total heating surface,	35·0	35·0	40·0	34·0	36·0	37·0	34	33	33	30·4	32·0	32·8	...	...	...	...
I.H.P. per ton gross register, - -	2·15	1·75	4·0	0·60	0·58	1·25	0·45	0·46	0·65	...	...	...	...	5·00	4·00	3·0
Fuel per I.H.P. per hour, - lbs.,	1·30	1·27	1·60	1·27	1·27	1·45	1·30	1·45	1·5	1·80	1·85	1·87	2·5	2·00	2·10	2·25
I.H.P. force draught per ton, - -	6·9	7·0	10·5	4·73	5·10	8·0	...	...	...	10·4	11·7	19·0	43·6	7·47	8·00	7·47
I.H.P. force draught per sq. ft. of grate,	18·0	18·0	20·0	16·0	16·0	18·0	...	...	...	13·4	12·3	20·3	26·0	15·0	13·0	10·0
I.H.P. force draught per 100 sq. ft. of total heating surface,	45·0	45·0	52·0	36·0	38·0	40·5	...	...	...	38·0	40	41·0	45·0	45·0	42·0	40·0
Ratio L.-P. to H.-P. cylinder, triple,	7·5	7·0	6·0	8·0	8·0	7·0	7·5	7·2	7·0	6·3	6·8	7·1	5·75	5·0	6·0	...
Do. do., quadruple,	9·0	8·3	...	8·8	8·7	8·0	8·9	8·3	8·0	...	...	...	...	...	...	...
Do. do., compound,	...	...	...	...	...	...	...	...	...	...	...	...	...	3·5-3·85	3·7-4·0	3·5-4·0

TABLE L.—PARTICULARS OF NAVAL SHIPS'

Designation.	When Built.	Number of Screws.	Kind of Engines.	Cylinders.			Full Speed.		
				Number.	Diameter.	Stroke.	Revolutions per Minute.	Piston Speed.	Referred Mean Pressure.
					Ins.	Ins.		Ft.	Lbs.
Battleship.	1870	1	Horizontal Trunk Surface-Condensing, -	2	127 (= 118)	54	71·5	643	20·1
Do.,	1870	1	Do. Return Connecting Rod Surface Condensing, - - - -	2	120	54	62·7	564	19·5
Do.,	1870	1	Do. do. do. do.,	2	96	48	68·4	547	19·6
Do.,	1870	2	Do. do. do. do.,	4	72	36	72·2	433	23·0
Do.,	1880	2	Vertical Direct-Acting Compound, - -	4	60-104	42	82·1	575	22·3
Do.,	1885	2	Do. do. do., - -	6	77-55-77	48	80·0	640	25·8
Do.,	1888	2	Do. do. do., - -	6	..	45	100·0	750	..
Do.,	1890	2	Do. do. Triple, - - - -	6	43-62-96	51	95·0	807	34·8
Do.,	1896	2	Do. do. do., - - - -	6	40-59-88	51	103·0	875	33·1
Do.,	1897	2	Do. do. do., - - - -	6	40-59-88	51	103·0	875	33·4
Do.,	1899	2	Do. do. do., - - - -	6	30-49-80	51	108·5	922	49·0
Do.,	1901	2	Do. do. do., - - - -	6	31½-51½-84	51	110·2	936	50·5
Do.,	1903	2	Do. do. do., - - - -	8	33½-54½-63-63	48	120·0	960	50·2
Cruisers— 1st Class,	1870	1	Horizontal Trunk Surface-Condensing, -	2	(= 104½)	48	74·1	593	24·0
Do.,	1877	2	Do. Return Connecting-Rod Compound,	3	41/75-41/75	36	97·2	553	24·8
Do.,	1885	2	Do. do. do. do.,	4	46-86	39	95·1	618	26·1
Do.,	1888	2	Do. Direct-Acting Triple, - - -	6	36-51-78	42	116·0	812	37·1
Do.,	1890	2	Vertical do. do., - - - -	12	36-52-80	48	105·0	840	41·8
Do.,	1894	2	Do. do. do., - - - -	6	40-59-88	51	106·0	900	33·4
Do.,	1897	2	Do. do. do., - - - -	8	45-70-76-76	48	113·0	904	51·7
Do.,	1899	2	Do. do. do., - - - -	8	34-55½-64-64	48	116·6	932	46·7
Do.,	1901	2	Do. do. do., - - - -	8	34-55½-64-64	48	118·9	951	50·3
Do.,	1902	2	Do. do. do., - - - -	8	36-59-68-68	48	120·0	960	50·6
Do.,	1902	2	Do. do. do., - - - -	8	43½-71-81½-81½	48	122·4	979	50·8
Do.,	1903	2	Do. do. do., - - - -	8	37-60-69-69	42	140·0	980	49·3
Do.,	1903	2	Do. do. do., - - - -	8	43½-69-77-77	42	135·0	935	44·1

TABLE La.—PARTICULARS OF NAVAL SHIPS'

Cruisers— 2nd Class,	1870	1	Horizontal Trunk Surface-Condensing, -	2	(=86)	45	79·1	593	21·5
Do.,	1870	1	Do. Direct-Acting Surface-Condensing,	2	88	42	74·2	520	20·3
Do.,	1877	1	Do Return Connecting-Rod Compound,	3	88-72-88	48	70·0	560	24·0
Do.,	1882	2	Do. Direct-Acting Compound, - -	4	42-80	45	90·0	675	26·7
Do.,	1886	2	Do. do. do., - -	4	33-64	39	120·0	780	40·4
Do.,	1889	2	Do. do. Triple, - - - -	6	34½-51-76½	36	140·0	820	41·6
Do.,	1889	2	Vertical do. do., - - - -	6	33½-49-74	39	140·0	910	40·0
Do.,	1891	2	Do. do. do., - - - -	6	33½-49·74	39	140·0	910	38·8
Do.,	1896	2	Do. do. do., - - - -	6	33-49-74	39	144·2	936	40·3
Do.,	1898	2	Do. do. do., - - - -	6	26-42-68	39	141·0	916	50·9
Do.,	1900	2	Do. do, do., - - - -	8	26-42-48-48	30	181·3	906	49·2
Cruisers— 3rd Class,	1870	1	Horizontal Return Connecting-Rod Surface Condensing, - - - -	3	55	30	90·3	451	20·0
Do.,	1875	1	Do. Return Connecting-Rod Compound,	2	57-90	33	95·8	527	21·1
Do.,	1878	1	Do. do. do. do.,	4	46-64	36	96·0	576	22·1
Do.,	1884	1	Do. do. do. do.,	4	42-72	36	80·3	481	26·3
Do.,	1885	2	Do. Direct-Acting Compound, - -	4	26-46	30	150·0	750	44·4
Do.,	1887	2	Do. do. do., - -	4	27-50	33	150·0	825	33·2
Do.,	1888	2	Do. do. Triple, - - - -	6	26-37-57	33	150·0	825	36·4
Do.,	1890	2	Vertical do. do., - - - -	6	26½-39-57	27	220·0	990	30·0
Do.,	1890	2	Do. do. do., - - - -	6	30½-45-68	33	160·0	880	24·0
Do.,	1891	2	Do. do. do., - - - -	6	30½-45-68	33	160·0	880	37·1
Do.,	1898	2	Do. do. do., - - - -	6	20½-33-54	27	318 0	981	52·7
Do.,	1901	2	Do. do. do., - - - -	8	20½-33 -33½-33½	24	250·0	1000	52·0
Do.,	1903	2	Do. do. do., - - - -	8	24½-33½-42½-42½	24	250·0	1000	57·3

MACHINERY (BATTLESHIPS AND 1ST CLASS CRUISERS).

Boilers.				Full Power Trial.		4/5 or Natural Draught.		Weight of Machinery.			I.H.P. Full Speed per			Weight + Displacement.
Number and Kind.	Pressure.	Grate.	Total Heating Surface.	I.H.P.	Speed	I.H.P.	Speed.	Total.	Full Power per I.H.P.	Natural Draught per I.H.P.	Ton of Machinery.	Square Foot of Grate.	100 Feet of Total Heating Surface.	
	Lbs.	Sq. ft.	Sq. ft.		Knots.		Knots.	Tons.	Lbs.	Lbs.				
Box,	30·0	840	22,200	8,539	14·69	7,187	13·8	1090	286	339	7·83	10·2	38·5	0·126
Do.,	31·6	..	..	7,842	14·97	7,470	14·7	1043	300	313	7·48	..	..	0·129
Do.,	30·0	570	14,388	4,960	13·75	..	..	714	327	..	6·86	8·6	34·0	0·103
Do.,	31·7	515	14,115	4,914	14·0	4,311	13·2	750	342	389	6·55	9·5	33·9	0·125
Cylindrical,	65	608	17,500	6,624	14·4	..	..	998	337	..	6·64	10·9	37·9	0·131
Oval and cylindrical,	90	667	22,784	10,184	16·75	8,000	16·2	1266	278	354	8·04	15·3	44·7	0·151
S.E. cylindrical,	90	790	20,587	11,271	16·75	8,277	16·0	1209	241	331	9·30	14·2	54·7	0·114
Cylindrical,	135	629	18,930	12,465	16·75	8,813	16·2	1015	189	258	12·20	19·7	65·8	0·015
S.E. cylindrical,	155	731	20,348	11,500	17·5	9,430	16·75	1163	224	276	10·00	16·0	56·6	0·082
8 do. do.,	155	817	25,233	12,414	18·0	10,404	17·0	1341	241	282	9·30	15·2	49·3	0·090
Belleville,	300	1050	33,770	13,763	18·5	10,454	17·2	1290	209	280	10·70	13·1	40·8	0·100
Do.,	300	1170	37,120	15,503	18·5	11,626	..	1400	202	270	11·10	13·2	41·8	0·093
Mixed,	..	1390	43,260	18,220	19·0	13,670	..	1580	196	260	11·40	13·1	42·1	0·113
Box,	30	900	24,868	7,361	16·5	..	..	1043	317	..	7·06	8·18	29·6	0·180
S.E. oval,	65	690	19,500	7,714	18·5	6,658	17·0	1012	294	340	7·62	11·18	39·5	0·271
8 S.E. cylindrical,	90	546	15,160	5,661	16·6	..	..	776	307	..	7·29	10·37	37·2	0·180
4 D.E. cylindrical,	130	492	15,190	8,738	18·75	6,090	18·0	772	198	279	11·30	17·7	57·5	0·138
D.E. cylindrical,	155	1135	31,078	21,411	21·6	14,924	20·4	1543	162	231	13·80	18·8	68·8	0·171
4 D.E. 1 S.E. cylindrical,	155	812	24,908	12,851	20·5	10,517	19·9	1161	202	247	11·10	15·2	51·4	0·158
Belleville,	300	2200	67,800	25,774	22·0	22,547	21·5	2230	195	224	11·50	11·7	38·0	0·157
Do.,	300	1460	40,990	16,961	20·5	12,785	19·8	1540	204	270	11·00	11·6	42·6	0·140
Do.,	300	1390	47,300	19,156	21·0	14,049	19·9	1577	185	250	12·10	13·8	40·5	0·143
Do.,	300	1650	51,600	21,400	21·75	16,440	..	1800	188	245	11·90	13·0	41·3	0·150
Do.,	300	2314	71,964	31,409	24·0	23,108	..	2500	178	242	12·60	13·6	43·6	0·178
Do.,	300	1610	50,300	22,360	22·7	16,100	..	1750	175	243	12·80	13·8	44·4	..
Mixed,	205	1760	62,250	23,500	23·0	..	..	2250	215	..	10·44	13·3	37·7	..

MACHINERY (2ND AND 3RD CLASS CRUISERS).

Box,	30	504	12,748	4,500	15·12	..	..	611	304	..	7·36	9·00	35·4	0·200
Do.,	30	455	12,420	3,878	14·82	..	..	640	370	..	6·06	8·52	31·3	0·209
S.E. cylindrical,	70	510	12,700	4,950	14·5	..	..	873	395	..	5·67	9·70	38·9	0·252
8 do. do.,	90	542	14,924	5,500	16·6	4,100	..	824	336	448	6·67	10·16	37·0	0·219
Gunboat,	110	388	11,565	6,151	18·0	4,217	..	559	204	300	11·0	15·8	53·0	0·157
4 D.E. cylindrical,	155	570	13,830	9,653	19 0	6,215	18·0	649	151	234	14·8	16·9	70·0	0·220
Do. do.,	155	525	12,682	9,435	19·0	6,144	18·0	624	148	228	15·3	17·9	74·4	0·221
Do. do.,	150	575	15,641	9,271	20·5	7,423	19·3	748	181	226	12·4	16·1	59·4	0·220
8 S.E. do.,	155	624	18,579	9,846	20·3	8,307	19·6	915	208	247	10·8	15·8	53·0	0 210
Belleville,	309	869	25,606	10,272	20·0	7,155	18·7	825	180	258	12·8	11·9	40·1	0·142
Do.,	300	792	24,426	9,770	20·5	7,660	19·3	832	190	243	11·7	12·3	40·0	0·149
Box,	30	210	5,980	2,170	12·8	..	..	236	244	..	9·19	10·3	36·3	0·152
6 S.E. cylindrical,	60	244	5,850	2,139	13·0	..	..	350	374	..	6·01	8·9	36·6	0·165
Gunboat,	60	246	6,312	2,400	13·0	..	..	394	365	..	..	..	38·1	0·165
S.E. cylindrical,	90	301	9,254	4,030	14·6	3,131	..	517	300	370	7·8	..	..	..
Gunboat,	120	218	6,400	3,365	16·5	2,201	..	296	197	301	11·3	15·8	52·6	0·207
Do.,	130	222	6,836	3,754	16·5	2,462	..	353	210	321	10·6	16·9	55·1	0·216
Do.,	140	244	7,878	4,613	17·5	2,647	..	395	192	334	11·6	18·9	58·4	0·242
Locomotive,	155	244	7,088	4,561	18·5	3,592	17·8	274	135	170	16·7	18·7	64·2	0·149
4 D.E. cylindrical,	155	365	9,938	..	..	4,615	17·2	515	..	249	..	..	46·5	0·200
Do. do.,	155	410	11,025	7,469	19·5	5,016	17·6	539	156	257	13·8	18·2	69·7	0·209
Express water-tube,	300	352	18,100	7,152	20·7	5,388	19·5	383	120	159	18·9	20·3	39·7	0·179
Do. do.,	300	352	17,760	7,331	20·8	5,218	19·5	370	113	159	19·8	20·8	41·7	0·168
Do. do.,	300	490	26,000	9,800	..	..	..	..	..	..	..	..	..	..

TABLE LI*b*.—PARTICULARS OF

Designation.	When Built.	Number of Screws.	Kind of Engines.	Cylinders.			Full Speed.		
				Number.	Diameter.	Stroke.	Revolutions per Minute.	Piston Speed.	Referred Mean Pressure.
					Ins.	Ins		Ft.	Lbs.
Gunboat, -	1876	1	Horizontal Return Connecting-Rod Compound, - - - - -	2	31-48	18	137	411	23·8
Do., -	1880	1	Horizontal Direct-Acting Triple, - -	3	20-30-45	24	184	736	36·6
Torpedo gun-boat,	1886	2	Vertical do. do., - -	6	22-33-49	18	310	930	..
Do.,	1890	2	Do. do. do., - -	6	22-33-49	21	250	875	38·4
Do.,	1890	2	Do. do. do., - -	6	24-34-51	21	250	875	34·2
Vidette boat,	..	1	Do. do. Compound, -	2	8½-13	9	460	690	54·8
Torpedo boat,	1877	1	Do. do. do., -	2	9½-15	9·5	435	689	47·6
Do.,	1877	1	Do. do. do., -	2	12¾-21	12	354	708	46·5
Do.,	1886	1	Do. do. do., -	2	18-30	16	402	1072	44·3
Do.,	1887	1	Do. do. Triple, - -	3	15-22-34	16	402	1072	36·4
Do.,	1894	1	Do. do. Quadruple, -	4	14-20-27-36	16	410	1093	47·2
Do.,	1893	1	Do. do. Triple, - -	3	18-26-39½	18	370	1110	55·8
Destroyer, -	1893	2	Do. do. do., - -	6	18-26-39½	18	360	1078	43·8
Do., -	1894	2	Do. do. do., - -	6	18-26-39½	18	393	1178	44·5
Do., -	1894	2	Do. do. do., - -	8	19-27-27-27	16	392	1046	60·0
Do., -	1894	2	Do. do. do., - -	6	19-29-43	18	366	1098	46·4
Do., -	1895	2	Do. do. do., - -	6	18-27½-42	18	366	1097	42·2
Do., -	1896	2	Do. do. do., - -	6	18-27-42	18	397	1190	43·7
Do., -	1897	2	Do. do. do., - -	8	20-29-30-30	18	395	1186	56·9
Do., -	do.,	2	Do. do. do., - -	6	21-32½-48	18	365	1095	53·2
Do., -	do.,	2	Do. do. do., - -	6	19-29½-46	18	395	1185	52·1
Do., -	do.,	2	Do. do. do., - -	8	20¾-31-34-34	18	392	1176	50·0
Sloop, - -	1899	1	Do. do. do., - -	6	16½-26½-42	24	200	800	40·0
Gunboat, -	1898	2	Do. do. do., - -	6	11-17-27	16	286	763	55·3
Sloop, - -	1901	2	Do. do do., - -	6	11½-18-29½	24	200	800	42·6

MACHINERY OF SMALL NAVAL SHIPS.

Boilers.				Full Power Trial.		4/5 or Natural Draught.		Weight of Machinery.			I.H.P. Full Speed per			Weight + Displacement.
Number and Kind.	Pressure.	Grate.	Total Heating Surface.	I.H.P.	Speed.	I.H.P.	Speed.	Total.	Full Power per I.H.P.	Natural Draught per I.H.P.	Ton of Machinery.	Square Foot of Grate.	100 Feet of Total Heating Surface.	
	Lbs.	Sq. ft.	Sq. ft.		Knots.		Knots.	Tons.	Lbs.	Lbs.				
2 Gunboat, - -	60	45	1,018	519	10·0	..	..	79·7	344	..	6·51	11·53	50·9	0·233
Do., - -	150	68	2,200	1298	14·1	1033	12·4	108·3	185	235	12·0	19·10	59·0	0·135
4 Locomotive, - -	150	121	4,426	2696	18·5	..	..	120·0	97	..	23·1	22·4	61·3	0·219
Do., - -	150	190	5,330	3840	20·0	2598	..	171·0	100	149	22·4	20·2	72·0	0·232
Do., - -	150	171	6,204	3665	19·0	2631	..	211·0	130	178	17·3	21·4	59·1	0·287
1 Locomotive, - -	146	5·5	235	151	15·0	..	..	4·6	68	..	33·0	27·5	64·3	0·514
Do., - -	130	7	278	175	15·2	..	..	5·51	70·4	..	31·8	25·0	63·0	0·441
Do., - -	120	15·5	630	400	17·5	..	..	11·0	61·5	..	36·4	26·0	63·5	0·410
Do., - -	130	41	1,775	1013	19·0	..	..	31·6	70	..	32·0	24·7	57·0	0·500
Do., - -	180	38	1,660	1057	19·5	..	..	31·5	66·9	..	33·5	27·8	63·7	0·500
2 Water-tube Yarrow,	250	50	2,350	1600	23·0	..	..	..	..	..	..	32·0	68·1	..
Do. do.,	220	75	3,500	2300	23·5	..	..	48·8	47·5	..	47·1	30·7	66·0	0·377
2 Locomotive, - -	180	100	4,656	3497	26·1	..	..	94·0	60	..	37·2	35·0	75·0	0·392
Water-tube Yarrow,	180	154	8,216	3884	27·6	..	..	90·6	52	..	42·8	25·1	47·3	0·377
Do. Thornycroft,	210	189	9,780	4368	28·4	..	..	108·2	56	..	40·3	23·1	44·7	0·438
Do. Normand, -	175	150	8,800	4484	27·3	..	..	123·2	62	..	36·3	29·9	50·9	0·440
Do. Reed, - -	210	209	10,164	3885	27·9	..	..	116·2	67	..	33·4	18·5	38·7	0·461
Do. Blechynden,	200	176	10,022	4367	27·4	..	..	106·0	54	..	41·1	24·8	43·6	0·400
Do. Thornycroft,	220	196	11,855	5795	30·1	..	..	127·0	49	..	45·6	29·5	48·9	0·462
Do. Normand, -	220	210	11,945	6347	30·1	..	..	143·5	50	..	44·2	30·2	53·0	0·478
Do. Reed, - -	250	255	13,384	6318	30·3	..	..	143·7	51	..	43·9	24·7	47·2	0·479
Do. Thornycroft,	250	245	13,561	6523	30·2	..	..	126·0	48	..	51·7	27·5	48·0	0·420
4 Belleville, - -	260	135	4,020	1400	13·2	1120	12·0	140·0	224	280	10·0	10·4	35·0	0·143
4 Yarrow, - -	250	80·5	4,000	1300	13·5	960	13·0	95·0	164	..	13·7	16·1	32·5	0·136
4 Babcock, - -	210	144	4.040	1404	13·5	1021	..	162·5	260	356	8·6	10·0	35·0	0·151

TABLE Lc.—PARTICULARS OF OCEAN EXPRESS STEAMERS' MACHINERY.

Name of Ship.	When Built.	Tonnage.		Number of Screws.	Cylinders.			Number of Revolutions.	Speed of Piston per Minute.	Referred Mean Pressure	Boilers.				Trial Trip.		Weight of Machinery		Trial I.H.P. per				Weight of Machinery. ÷ Displacement.
		Gross Register.	Load Displacement.		Number.	Diameter.	Stroke.				Number and Kind.	Pressure.	Grate.	Total Heating Surface.	Total I.H.P.	Speed.	Total.	Per I.H.P.	Ton of Machinery.	Ton Gross Register.	Square Foot of Grate	100 Feet of Total Heating Surface.	
						Ins.	In.		Ft.	Lbs.		Lbs.	Sq. ft.	Sq. ft.		Kts.	Tons.	Lbs.					
Britannic, - -	1874	5,008	8,500	1	4	48/83-48/83	60	57·5	575	28·7	8 D.E. cylindrical,	70	680	19,090	5,400	16·0	1112	461	4·86	1·08	7·94	28·2	·131
Gallia, - -	1878	4,809	8,600	1	3	80-63-80	60	56·0	560	31·0	8 S.E. do.,	75	538	13,860	5,300	15·9	1116	473	4·75	1·10	9·85	38·2	·130
Arizona, - -	1879	5,160	11,600	1	3	90-62-90	66	55·0	605	27·0	S. and D.E. do.,	90	780	19,500	6,300	17·0	1335	474	4·72	1·22	8·08	32·3	·115
Ravenna, - -	1880	3,500	4,900	1	2	54-94	60	60·5	605	26·6	6 S.E. do.,	75	336	10,932	3,346	14·7	..	..	..	0·97	10·10	31·1	..
Servia, - -	1881	7,392	14,500	1	3	100-72-100	78	53·0	689	31·5	D. and S.E. oval,	90	1014	27,453	10,300	16·9	1877	408	5·49	1·40	10·16	37·4	·129
Alaska, - -	1881	6,932	13,000	1	3	100-68-100	72	64·2	770	29·1	Cylindrical, - -	100	1246	35,236	10,680	18·0	1860	389	5·74	1·54	8·57	30·3	·143
Aurania, - -	1883	7,269	14,000	1	3	91-68-91	72	58·0	696	29·1	Cylindrical, - -	90	1008	23,777	8,000	17·4	1728	484	5·00	1·10	8·0	33·6	·123
Oregon, - -	1883	7,353	13,700	1	3	104-70-104	72	65·3	784	33·0	Do., - -	110	1575	38,062	13,300	18·3	2205	371	6·03	1·80	8·44	35·0	·161
Umbria, - -	1884	8,128	15,500	1	3	105-71-105	72	67·5	810	33·6	Do., - -	110	1680	38,817	14,300	19·0	2330	365	6·13	1·76	8·51	39·4	·150
Oroya, - -	1886	6,297	10,500	1	3	40-66-100	72	64·5	774	36·6	6 D.E. cylindrical,	160	626	17,640	6,751	16·5	1343	446	5·03	1·07	10·8	38·4	·128
Victoria, - -	1887	6,527	8,124	1	3	40½-63½-102	72	68·0	756	33·9	Cylindrical, - -	150	632	15,610	6,347	17·0	..	..	..	0·97	10 0	40·7	..
New York, -	1888	10,500	18,000	2	6	45-71-113	60	..	..	..	Do., - -	150	1293 F	50.040	18,400	20·5	..	..	..	1·75	14·2	36·8	..
Teutonic, - -	1889	9,686	18,000	2	6	43-68-110	60	78·0	780	37·8	Do., - -	180	1154 F	40,972	17,000	21·0	2778	366	6·12	1·76	14·7	41·5	·103
Oruba, - -	1889	5,857	9,700	1	3	39-66-97	66	67·8	746	33·5	Do., - -	160	530	15,107	5,525	16·5	1257	510	4·40	0·94	10·4	36·6	·130
F. Bismark, -	1890	8,430	15,300	2	6	43½-67-106½	63	81·0	850	35·1	Do., - -	155	1450	47,000	16,000	20·5	..	..	..	1·66	11·0	34·0	..
Empress of India,	1891	5,905	11,500	2	6	32-54-82	54	88·0	792	39·9	4 D.E. cylindrical,	160	520 F	20,198	10,125	18·0	1414	313	7·16	1·71	19·4	49·9	·123
Ophir, - -	1891	6,900	10,600	Twin	6	34-51½-85	54	87·5	768	31·2	{ 5 D.E. & 2 S.E. cylindrical, }	160	756	26,004	8,386	18·5	..	..	..	1·21	11·09	32·2	..
Scot, - - -	1891	7,815	14,500	2	6	34½-57½-92	60	80·0	800	36·0	6 do. do.,	170	837 F	22,964	11,656	18·5	1720	330	6·78	1·50	13·9	50·7	·112
Australia, - .	1892	7,376	11,500	1	3	42½-67-108	72	..	..	..	{ 3 D.E. & 3 S.E. cylindrical, }	170	495 F	20,665	8,500	18·0	..	..	..	1·15	17·2	41·1	..
Campania, - -	1893	12,950	22,000	2	10	37/98-79-37/98	69	80·9	930	33·0	12 D.E. cylindrical,	165	2630	82,200	29,600	22·0	..	..	..	2·28	11·2	36·0	..
K. Wilhelm, D.G.,	1897	14,349	23,500	2	8	52-89·7-96·4-96·4	69	77·2	887	38·5	{ 12 D.E. & 2 S.E. cylindrical, }	178	2618	84,285	30,000	22·5	4460	333	6·73	2·09	11·5	35·6	·190
Oceanic, - -	1899	17,274	25,900	2	8	47½-79-93-93	72	79·0	948	35·8	15 D.E. cylindrical,	192	1962 F	74,686	28,000	21 0	4414	353	6·34	1·54	13·5	35·5	·170
Savoie, - -	1900	11,638	15,400	2	8	44½-68½-80-80	67	92·0	1027	36·5	16 S.E. do.,	175	1225 F	45,565	23,000	22·6	2506	244	9·18	2·00	18·8	50·4	·163
Deutschland, -	1900	16,503	24,500	2	12	104-$\frac{86·6}{106}$-$\frac{86·6}{106}$-74	73	..	..	..	{ 12 D.E. & 4 S.E. cylindrical, }	220	2188 F	85,468	33,900	23·5	5670	336	6·36	2·36	17·9	45·6	·231
K. Wilhelm II.,	1902	20,000	27,600	2	16	37/49-75-112	71	80·0	947	35·0	{ 12 D.E. & 7 S.E. cylindrical, }	225	3121	107,648	40,000	23·3	..	..	..	2·00	12·8	37·2	..
Kinfauns Castle,	1899	9,664	15,000	2	8	28-39½-57½-82	60	..	..	..	{ 4 D.E. & 2 S.E. cylindrical, }	210	880 F	31,187	12,220	18·2	..	..	..	1·26	13·9	39·2	..

TABLE *Ld.*—PARTICULARS OF THE MACHINERY OF CARGO STEAMERS TAKING PASSENGERS.

Name of Ship.	When Built.	Tonnage.		Number of Screws.	Cylinders.			Number of Revolutions.	Speed of Piston per Minute.	Referred Mean Pressure.	Boilers.				Trial Trip.		Weight of Machinery		Trial I.H.P. per				Weight of Machinery. + Displacement.
		Gross Register.	Load Displacement.		Number.	Diameter.	Stroke.				Number and Kind.	Pressure.	Grate.	Total Heating Surface.	Total I.H.P.	Speed.	Total.	Per I.H.P.	Ton of Machinery.	Ton Gross Register.	Square Foot of Grate.	100 Feet of Total Heating Surface.	
						Ins	In.		Ft.	Lbs.		Lbs.	Sq. ft.	Sq. ft.		Kts.	Tons.	Lbs.					
Salier, - -	1875	3,098	4,698	1	2	52-96	48	59·5	476	20·8	6 S.E. cylindrical,	60	312	8,340	2,180	13·0	460	472	4·74	0·701	6·99	26·1	·098
Eduardo, - -	1878	2,808	5,300	1	2	36-72	48	60·0	480	21·6	2 D.E. do.,	75	170	4,900	1,399	11·5	307	494	4·54	0·608	8·20	28·4	·058
A Monarch, -	1880	4,017	7,300	1	2	46-87	57	57·0	542	23·6	3 D.E. oval, - -	80	247	7,803	2,302	12·2	481	468	4·79	0·578	9·32	29·5	·067
Galileo, - -	1880	3,008	6,800	1	2	34-68	63	57·0	599	21·3	2 D.E., 1 S.E. cylindrical,	80	188	5,080	1,396	11·0	320	514	4·36	0·465	7·42	27·4	·053
Pomeranian, -	1881	4,258	8,500	1	2	43-86	60	57·4	574	23·7	2 D.E., 2 S.E. cylindrical,	80	234	8,655	2,389	12·5	539	505	4·43	0·561	10·21	27·6	·063
Rosario, - -	1883	1,862	3,600	1	3	20½-33-58	36	62·0	372	22·3	2 S.E. cylindrical,	150	57	2,170	664	10·0	190	641	3·50	0·357	11·65	30·6	·053
Martello, - -	1884	3,721	7,800	1	3	31-50-82	57	59·0	561	26·2	2 D.E., 2 S.E. cylindrical,	150	192	7,000	2,320	12·0	528	505	4·40	0·624	12·14	33·1	·068
Apollo, - -	1886	3,181	6,900	1	3	25-40-66	48	73·0	584	24·8	2 D.E. cylindrical,	160	128	4,760	1,502	11·0	313	466	4·80	0·469	11·74	31·5	·045
Cufic, - - -	1888	4,827	10,600	1	3	27-44½-74	60	67·0	670	33·3	Cylindrical, - -	..	216	7,392	2,900	13·0	533	412	5·44	0·600	13·43	39·2	·050
Cazengo, - -	1889	2,889	5,180	1	3	32-48-80	48	73·0	584	34·2	2 D.E. cylindrical,	160	192 H	6,750	3,036	14 0	508	373	6·00	1·050	15·80	44·9	·098
Bovic, - -	1892	6,583	14,500	2	6	22½-36½-60	48	80·0	640	34·5	Cylindrical, - -	..	346	11,512	3,800	13·0	796	481	4·66	0·577	10·98	33·0	·055
Kensington, -	1894	8,669	17,850	2	8	25½-37½-52½-74	54	87·0	783	40·0	S.E. cylindrical, -	200	382 F	18,549	8,800	15·8	1800	352	6·38	0·968	21·73	44·8	·073
Georgic, - -	1895	10,077	20,000	2	6	24-39½-65½	51	77·0	655	34·5	Cylindrical, - -	..	394	13,298	4,610	13·0	840	408	5·49	0·456	11·70	34·7	·042
K. Wilhelmina, -	1896	4,248	6,575	1	4	27-39-56-80	54	68·2	614	36·0	D.E. cylindrical, -	210	344	11,660	3,350	14·0	754	504	4·44	0·788	9·74	28·6	·118
Othello, - -	1896	5,059	11,200	1	3	24¾-40-70	48	71·0	568	36·7	3 S.E. cylindrical,	200	182	6,414	2,430	11·0	434	400	5·60	0·456	11·46	32·6	·039
Cymric, - -	1898	12,647	25,000	2	8	25½-36½-53-75½	54	81·0	729	36·9	Cylindrical, - -	210	593	21,100	7,810	15·0	1528	468	4·78	0·580	12·33	34·7	·061
K. Wilhelm I., -	1898	4,446	7,845	1	4	27½-39-55-82	60	73·0	730	32·6	D.E. cylindrical, -	210	253 F	9,759	3,810	14·5	769	452	4·95	0·847	15·10	38·5	·098
Cleopatra, - -	1899	6,849	13,900	1	3	32-54-90	66	65·0	715	34·4	2 D.E., 2 S.E. cylindrical,	200	306 F	13,080	4,500	14·5	897	446	5·02	0·657	14·70	34·4	·065
Saxonia, - -	1900	13,963	22,580	2	8	29-41½-59-84	54	78·0	702	39·0	S.E. cylindrical, -	210	570 F	25,808	9,090	16·0	1985	490	4·58	0·651	15·90	35·3	·088
Runic, - -	1900	12,482	24,600	2	8	22-31½-46-67	51	81·0	689	35·3	Cylindrical, - -	210	485	17,100	5,200	13·0	1185	510	4·39	0·400	10·72	30·4	·047
Somali, - -	1901	6,760	11,700	2	6	23½-36½-60	48	..	..	..	2 D.E. cylindrical,	170	239	9,230	..	13·0	..	..	..	..	..	..	..
Celtic, - -	1901	20,880	30,000	2	8	33-47½-68½-98	63	80·0	840	35·2	8 do. do.,	210	1014	41,680	13,500	17·0	2975	493	4·54	0·600	12·33	30·0	·099
Missouri, - -	1902	..	17,200	2	6	25-42½-72	48	70·0	560	35·0	2 D.E., 2 S.E. cylindrical,	200	314 F	12,500	4,800	12·0	..	..	..	..	15·30	38·4	..

TABLE Le.—PARTICULARS OF SMALL PASSENGER STEAMERS' MACHINERY (SCREW).

Name of Ship.	When Built.	Tonnage.		Number of Screws.	Cylinders.			Number of Revolutions.	Speed of Piston per Minute.	Referred Mean Pressure.	Boilers.				Trial Trip.		Weight of Machinery		Trial I.H.P. per				Weight of Machinery. + Displacement.
		Gross Register.	Load Displacement.		Number.	Diameter.	Stroke.				Number and Kind.	Pressure.	Grate.	Total Heating Surface.	Total I.H.P.	Speed.	Total.	Per I.H.P.	Ton of Machinery.	Ton Gross Register.	Square Foot of Grate.	100 Feet of Total Heating Surface.	
						Ins.	In.		Ft.	Lbs.		Lbs.	Sq. ft.	Sq. ft.		Kts.	Tons.	Lbs.					
Cameo, - -	1876	1244	2400	1	2	33½-64½	42	61·0	427	23·1	2 S.E. cylindrical,	70	108	3,357	979	12·0	219	500	4·49	0·768	9·07	29·1	·091
Folkestone, -	1878	420	810	2	4	20-39	24	118·0	472	17·5	Do. do.,	70	75	2,100	601	13·0	129	480	4·66	1·43	8·00	23·6	·159
Edith, - -	1879	560	1100	2	4	21-40	27	113·0	508	20·0	Do. do.,	75	77	2,514	774	13·0	143	415	5·41	1·38	10·06	30·8	·130
Albert, - -	1880	536	1040	1	2	27-53	33	84·0	462	23·7	Do. oval, - -	80	67	2,200	733	12·5	132	406	5·55	1·37	10·94	33·3	·127
Lincoln, - -	1882	1075	2300	1	2	36-69	39	71·0	462	26·5	Do. cylindrical,	85	100	3,450	1390	12·7	233	376	6·00	1·29	13·90	40·3	·101
Retford, - -	1882	951	1600	2	4	23-44	27	108·0	486	23·4	Do. do.,	85	83	2,700	1065	13·0	174	366	6·12	1·12	12·83	39·4	·109
Norwich, - -	1883	1117	1750	2	4	30-57	36	90·0	540	22·4	2 D.E. do.,	80	220	5,880	2142	14·0	384	389	5·58	1·91	9·74	36·3	·219
Cambridge, -	1885	1259	2000	2	4	30-57	36	89·0	534	23·8	Do. do.,	85	220	5,820	2260	14·6	386	382	5·86	1·72	10·27	38·9	·193
Eldorado, - -	1886	1514	2200	1	3	28-43-70	39	88·0	572	31·2	Do. do.,	154	140 H	5,620	2082	15·0	296	315	7·10	1·45	14·87	37·1	·133
Juno, - - -	1888	1080	1650	1	3	20½-32-54	33	101·0	555	31·2	2 S.E. do.,	160	74 H	2,960	1200	13·0	182	339	6·60	1·11	16·20	40·5	·110
Gazelle, - -	1889	672	790	2	6	16½-26-41	30	121·0	605	34·1	Do. do.,	150	118 A	3,800	1650	17·0	250	339	6·60	2·45	14·00	43·4	·372
Blarney, - -	1889	1250	2270	1	3	24-38-62	42	83·0	581	32·5	2 D.E. do.,	160	126	4,080	1550	13·0	289	418	5·36	1·24	12·30	38·0	·123
Liberty, - -	1890	895	1710	1	3	24-39-64	33	100·0	550	33·4	2 S.E. do.,	160	132	4,500	1786	13·5	250	311	7·14	2·00	13·50	39·7	·146
Lydia, - -	1890	1059	1870	2	6	24-37-56	33	165·0	907	44·0	2 D.E. do.,	160	320 F	11,956	5977	19·5	530	200	11·09	5·64	18·7	49·0	·283
Duke of Clarence,	1892	1458	1780	2	6	22-34-51	33	150·0	825	39·8	Do. do.,	160	258 F	8,000	4000	18·5	420	235	9·51	2·75	15·5	50·0	·236
Bruno, - -	1892	841	1775	1	3	23½-35-57	33	104·0	572	34·6	2 S.E. do.,	150	114	3,920	1530	14·2	192	281	7·97	1·82	10·20	39·2	·108
Chelmsford, -	1893	1635	2250	2	6	26-39½-61	36	130·6	780	31·8	5 do. do.,	160	357	11,450	4400	18·0	593	302	7·42	2·69	12·32	38·4	·264
Ibex, - - -	1895	1062	1270	2	6	22-34-51	33	152·0	836	41·0	2 D.E. do.,	160	250 F	8,000	4250	19·0	420	221	10·12	4·00	17·0	53·1	·331
Vienna, - -	1895	1753	2342		6	26-39½-61	36	184·0	804	38·2	5 S.E. do.,	160	357 A	11,200	5450	18·0	587	241	9·29	3·11	15·27	48·6	·251
Prince Edward, -	1897	1414	1260	2	6	19-30-48	24	192·0	768	36·7	2 D.E. do.,	190	170 F	6,200	3100	18·8	304	220	10·20	2·19	18·23	50·0	·217
Roebuck, - -	1897	1186	1740	2	6	23-36-56	33	160·0	880	47·2	Do. do.,	175	250 F	10,850	6200	20·0	..	..	..	5·23	24·80	57·4	..
Munster, - -	1897	2632	2850	2	8	29-45-48-48	33	175	962	..	4 do. do.,	..	520 F	18,500	9600	23·8	850	200	11·17	3·61	18·5	51·3	·309
Duke of Cornwall,	1898	1540	2120	2	8	22½-34-33½-33½	33	160·0	880	44·4	4 S.E. do.,	180	317 F	10,772	5520	19·8	612	248	9·02	3·59	17·40	51·1	·249
Prince George, -	1899	2040	2500	2	8	26-40-45-45	30	177·0	885	36·2	Do. do.,	180	280 F	11,440	6000	20·3	568	212	10·56	2·80	21·70	52·6	·227
Anglia, - -	1901	1862	2300	2	8	26-40-43-43	33	175	962	..	8 do. do.,	..	480 F	15,000	7170	21·0	..	..	..	..	..	..	..
Great Western, -	1902	1339	..	2	8	19-29½-33-33	30	..	..	..	2 D.E. cylindrical,	180	209 F	7,492	3600	17·0	..	..	..	..	..	..	..

TABLE LI.—PARTICULARS OF PADDLE-WHEEL STEAMERS' MACHINERY.

Name.	When Built.	Tonnage.		Kind of Engine.	Cylinders.			Number of Revolutions.	Speed of Piston.	Referred Mean Pressure.	Boilers.				Trial Trip.		Weight of Machinery		Trial I.H.P. per				Weight of Machinery. ÷ Displacement.
		Gross Register.	Displacement.		Number.	Diameter.	Stroke.				Number and Kind.	Pressure.	Grate.	Total Heating Surface.	Total I.H.P.	Speed.	Total.	Per I.H.P.	Ton of Machinery.	Ton Gross Register.	Square Foot of Grate.	100 Feet of Total Heating Surface.	
						Ins.	Ins.		Ft.	Lbs.		Lbs.	Sq. ft.	Sq. ft.		Kts.	Tons.	Lbs.					
Mirimaree, -	1871	..	..	Oscillating, -	2	80	84	29·0	406	23·8	Box, - - -	30	450	12,500	3,248	18·0	495	341	6·56	..	7·22	25·9	..
Osborne, - -	1872	..	..	Do., -	2	80½	84	28·6	400	25·0	Do., - - -	30	412	11,000	3,373	16·6	516	342	6·54	..	8·18	30·7	..
Mary Powell, -	1872	..	800	Beam, - -	1	72	144	23·0	552	22·0	....	30	152	2,660	1,490	17·2	..	..	..	..	9·80	56·0	..
Vigilant, - -	1872	..	..	Oscillating, -	2	64	54	35·0	315	30·8	Box, - - -	30	225	6,032	1,750	14·5	245	314	7·14	..	7·78	29·1	..
Bosphorus, -	1874	..	299	Compound Oscil.,	2	34-57	42	51·0	357	23·5	2 S.E. cylindrical,	60	78	1,928	650	14·0	121	417	5·37	..	8·33	33·6	·410
Scutari, - -	1875	..	..	Do. Diag.,	2	28-52	42	40·0	280	34·7	1 do. do.,	100	62	1,500	625	13·1	96	345	6·50	..	10·10	41·7	..
Bramapootra, -	..	..	..	Do. do.,	2	22-37½	30	56·0	280	30·9	2 Locomotive, -	90	40	1,068	288	12·5	38·3	298	7·57	..	7·20	27·2	..
South America, -	..	..	..	Do. Oscil.,	2	46-82	60	31·6	316	27·9	4 S.E. oval, - -	70	160	4,310	1,410	12·7	269	427	5·24	..	8·81	32·7	..
Russia, - -	..	..	..	Do. Diag.,	2	35-62½	51	34·0	289	27·6	2 do. do., - -	70	86	2,328	753	12·8	137	407	5·50	..	8·75	32·6	..
P. Marie, - -	1877	1606	..	Do. Oscil.,	2	60-104	84	32·7	458	30·0	4 do. cylindrical,	70	364	7,304	3,543	17·1	..	..	..	2·21	9·74	48·5	..
Invicta, - -	1882	1236	..	Oscillating, -	2	80	78	33·8	439	29·5	4 Box, - - -	34	604	14,890	3,967	17·7	522	295	7·60	3·21	6·57	27·5	..
Ireland, - -	1885	1952	..	Do., -	2	102	102	27·1	461	30·6	8 do., - - -	30	807	22,750	7,000	23·0	1050	336	6·67	3·60	8·75	30·8	..
Empress, - -	1887	1218	..	Compound Diag.,	2	58-104	72	45·3	544	43·1	4 D.E. cylindrical,	110	407 F	18,720	6,000	19·5	..	..	..	4·94	14·74	43·8	..
Grimsby, - -	1889	851	402	Do. do.,	2	35-60	48	47·0	376	30·5	3 Gunboat, - -	80	102 F	2,700	1,105	15·0	148	300	7·47	3·15	10·83	40·9	·369
Violet, - -	1890	1175	1485	Triple Stpl., -	3	44-70-108	78	35·0	455	32·2	6 Locomotive, -	150	282 F	10,022	4,070	19·5	550	302	7·42	3·47	14·43	40·7	·370
Neptune, - -	1892	378	..	Compound Diag.,	2	33-62	60	57·0	570	38·5	2 Gunboat, - -	115	..	..	2,000	18·5	..	..	..	5·00	..	..	..
Marie Henriette,	1893	1451	1850	Do. do.,	..	60-108	84	52·9	740	39·6	8 S.E. cylindrical,	117	465 F	14,100	8,130	22·2	..	..	..	..	..	..	..
La Marguerite, -	1894	2205	1868	Do. do.,	2	56-110	72	52·0	624	44·2	4 D.E. do.,	140	616 F	..	7,500	22·3	..	..	..	3·40	12·18	..	..
Priscilla, - -	1894	5292	5200	Do. do.,	4	51-95	132	24·5	539	40·4	10 S.E. do.,	150	850	34,700	9,345	19·0	..	..	..	1·76	11·00	27·2	..
Duchess of York,	1896	996	..	Do. do.,	3	68-48-68	72	43·0	516	37·8	4 do. do.,	120	270 F	8,416	4,300	17·0	455	237	9·45	4·31	15·91	51·2	..
Paris, - - -	1896	2000	..	Triple do.,	3	30-49-75	84	32·0	448	36·0	4 do. do.,	..	140 F	5,150	2,200	16·0	..	..	..	1·10	15·70	42·7	..
Essex, - -	1896	297	330	Compound do.,	2	24-48	42	55·0	385	36·9	2 do. do.,	110	81 F	2,260	780	14·0	104	298	7·50	2·61	9·63	34·2	·315
Alicia, - -	1896	230	260	Do. do.,	2	20-40	48	50·0	400	31·3	2 Locomotive, -	130	..	2,400	600	12·0	76	283	7·90	2·61	..	25·0	·304
Le Nord, - -	1898	1700	1850	Triple do.,	3	42-59-87	88	52·0	763	51·0	Water-tube, -	186	516 F	16,146	7,000	21·5	..	..	..	5·34	13·56	43·2	..
Empress Queen,	1897	2140	2500	Compound do.,	3	92-68-92	84	50·0	700	40·6	4 D.E. cylindrical,	140	652	26,000	11,440	21·7	..	..	..	5·32	17·5	44·0	..

CHAPTER XXIV.

WEIGHT—INERTIA AND MOMENTUM—BALANCING.

HITHERTO in this book little or no consideration has been given to the effect of the weight of any parts of the engine, although that of each one has some on itself and on its immediate surroundings, though generally it may be very trifling. The effect of the weight of some of the parts, however, is perceptible throughout the whole engine, and sometimes even makes its presence known from stem to stern of the ship. In the case of the propellers, both screw and paddle, their weight has been taken account of in estimating the stresses on their shafts and the pressure on the bearings supporting them; also the weight of the shafts themselves is not overlooked in considering the dimensions of their journals, bearings, &c. But in calculating the sizes of shafts and propellers no reference has been made to the fact that their weight does to some extent modify the stresses borne by their parts. As a rule, however, the effect of mere weight is so very small compared with that due to the steam pressure on the pistons, cylinder covers, &c., that in ordinary engines it is and may be generally neglected. But the great increase in rate of revolutions in modern engines, and the demand for greater perfection in working, as well as the absolute necessity for it in many cases, have drawn more particular attention to the subject; some most interesting investigations have been made by scientists, and from time to time methods devised whereby the ill effects due to the inertia of the moving parts of an engine may be modified or removed altogether, so that it may be run at any speed without distress of any kind to itself or surroundings, and with perfect freedom from vibration.

Fixed Parts—that is, those parts the centre of gravity of which have no motion but that due to the motion of the ship—may first be taken into consideration. The effect of their weight is simply that due to gravity, and acts always vertically. For example, the cylinder and its appurtenances of a vertical engine rest on the columns, and by their weight increase the load on the columns; on the other hand, the stress on the bolts connecting them to the columns is decreased by it. When, however, the ship is in motion they have momentum, so that in case of a sudden stoppage by grounding or collision the tendency would be for them to move forward and shear the bolts, &c., and to bend the columns as well as strain the column bases and connections to the ship. The tendency is also of the same nature when the ship rolls heavily, but in this case motion is transverse to the forward movement. It is usual and advisable in all vertical engines, especially in tall ones, to provide for these contingencies by fitting ties of a more or less flexible nature from the cylinders to a suitable part of the ship. In a minor degree the same remarks are applicable to the columns, especially if of cast iron and heavy, but being lower down and sharing in the external support given to the cylinders by the ties, there is no need to do more than brace them together, as is generally done when the columns are of wrought iron or steel.

By the above definition the shafting is a "fixed part," whose weight sets up in itself bending moments as well as shearing forces which may be sufficiently large to take into account with the twisting moment if the bearings are far apart. A rotating shaft has also critical speeds of whipping, at which the centrifugal forces become unbalanced; and although in the case of marine shafts dangerous stresses may not necessarily be set up, the result is a considerable amount of vibration and racking on the bearings. As a rule, greater stability is secured at speeds above the critical speed, but in marine practice it is perhaps best, if possible, to arrange for the critical speed to be above the maximum engine speed.*

Moving Parts.—The effect of weight in these, however, demands the chief attention, and in quick running engines the greatest attention of the designer, not so much, as will be seen, for its static as for its kinetic effect. Some of these have simple rectilineal motion, others have circular motion, while a few have a combination of both, and move in an elliptic orbit; others, again, reciprocate on curved lines. The static effect of the weight of the piston, piston-rod, crosshead, and connecting-rod, is to increase the load on the piston on the downstroke and decrease it on the upstroke, in both cases uniformly. The effect of the weight of the crank-arms and pin (if unbalanced) is also to increase the down and decrease the up load, but not uniformly in this case. The effect of the former is generally measured by dividing the weight in pounds by the area of the piston in square inches, and adding the result to the effective mean pressures of the steam on the top of the piston and deducting it from that on the bottom. This is shown graphically by drawing the diagram of effective pressures of the top above a reference line and that of the bottom below it: a new reference line is now drawn below the original at a distance representing the pounds thus ascertained and the ordinates measured from it used to determine the load on rods, &c., and the twisting moments on the crank-shaft.

Fig. 197 is the indicator diagrams from the high-pressure cylinder of a triple-compound engine 31 inches diameter; the weight of the piston, piston-rods, connecting-rods, &c., is 5728 lbs.; dividing this by 755 the area of the piston in square inches, the constant downward pressure due

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* The sag or deflection of a shaft whose length between supports is L feet, its weight per foot of length w pounds, and its radius r feet, is

$$\delta = \cdot 0166 \times \frac{w}{E}\left(\frac{L}{r}\right)^4,$$ E being the modulus of elasticity, 4·32 × 10^9 lbs. per square foot.

And, if N be the critical speed of whipping,

$$N = \sqrt{\frac{115 \cdot 5}{\delta}}$$ for shaft *supported* on bearings.

Example.—A shaft 10 inches (·834 ft.) diameter is supported on bearings 25 feet apart; to find the sag

$$\delta = \cdot 0166 \times \frac{264 \times 1 \cdot 296 \times 10^7}{4 \cdot 32 \times 10^9} = \cdot 0131 \text{ foot, and } N = \sqrt{\frac{115 \cdot 5}{\delta}} = \sqrt{\frac{115 \cdot 5}{\cdot 0131}} = 93 \cdot 4 \text{ revolutions per minute.}$$

If the shaft be regarded as *fixed* in the bearings,

N = 2·28 times the above value (δ being taken for supported shaft as before);

that is, in the example, N = 213 revolutions per minute.

For more than two equidistant bearings, the above formulæ still hold, but for bearings unequally spaced the problem is complicated These formulæ will generally indicate what speeds to avoid. For instance, we should conclude in the above example that the bearings are too far apart. The advanced mathematical reader may consult Dr. Chree on "Whirling of Shafts," *Proc. Physical Soc.*, vol. xix.

to the weight is equal to 7·6 lbs. per square inch. The figure in plain lines is that taken from the top, that in the dotted lines from the bottom of the cylinder. A B is taken as a base line to divide into any number of parts at points *b b′*, &c., through which ordinates *b e*, *b′ e′*, &c., may be

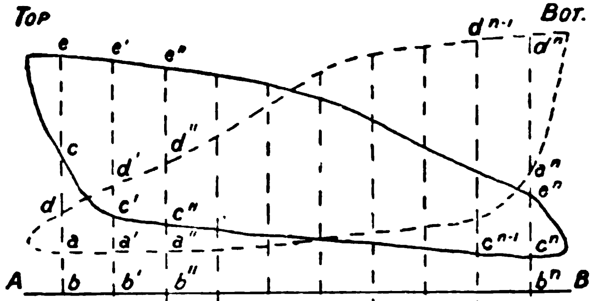

Fig. 197.—High-pressure Cylinder Diagram.

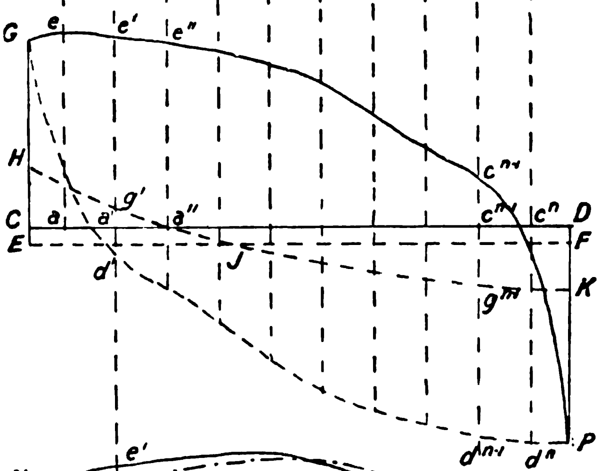

Fig. 198.—Diagram of Effective Pressures Neglecting Inertia.

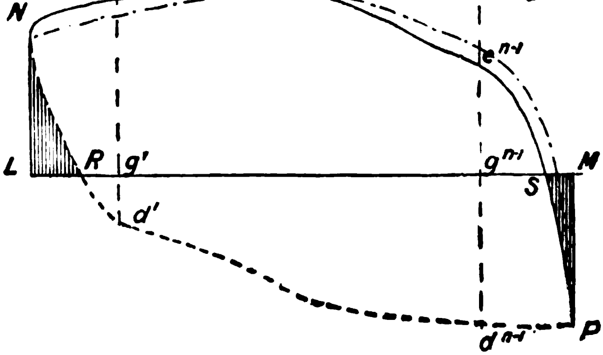

Fig. 199.—Diagram of Effective Pressures including those due to Inertia.

drawn, and of which *a e*, *a′ e′*, &c., represent the effective pressure on the top of the piston when in those positions on the downstrokes, and *c d*, *c′ d′*, &c., the effective pressures on the underside of the upstroke.

Now take CD as a base line (fig. 198) of the same length as AB; divide in the same way and draw a diagram by taking ordinates through the points and cutting off *a e*, *a' e'*, &c., same length as *a e*, *a' e'*, &c., in fig. 197, &c. On the lower side construct in a similar way the figure P *d'' d'* G from the dotted diagram, fig. 198.

Now take a point E below C, so that CE represents on the same scale as the diagrams the pressure per square inch on the piston equivalent to the weight of the piston, &c., and draw EF parallel to CD. Then any ordinates on EF, intercepted by G *e e'* P, &c., represent the true effective pressure on the downstroke, and similarly any ordinates from EF downward, intercepted by P *d' g* represent the true effective pressure on the upstroke at those points. The maximum ordinate on the lower side indicates the tension on the connecting-rod and piston-rod bolts for purposes of calculation, and the maximum ordinate on the top side of EF, the load on the connecting-rod, &c., for the same purpose; the actual load and tension being obtained, of course, by multiplying the number of pounds which these ordinates represent by the number of square inches of piston area. The curve of twisting moments of a vertical engine should also be obtained by using these ordinates.

A further modification may be made in the case of engines having the air and circulating pumps worked by means of levers, as then the weight of the moving parts of the pumps and their loads tend to balance the weight of the piston. Balance weights on the cranks opposite the arms have also been fitted with the object of balancing the *weights* of pistons, &c., and have also very properly been fitted with the sole object of balancing the crank weight only.

The weight of the slide valves and valve motions also modify in a similar way the stresses on their parts but in a minor degree.

In horizontal engines the pistons, rods, &c., do not by gravity affect the load on the pistons, but the weight of the crank-arms and part of the weight of the connecting-rod do affect the turning moment, but in a different way from that of a vertical engine. In their case the forward effort of the piston was resisted by the weight of crank, &c., till a little past the half-stroke and assisted from that position to the end of the stroke; on the back-stroke the assistance continued till nearly half-stroke when it ceased, and resistance set up and increased to the end. Here balance weights could be, and were, used to advantage, so that most of the horizontal engines had them, while a vertical engine was rarely seen with these appendages.

Momentum.—After all is said, the effect of mere weight or gravity on the parts of an engine is very limited and never serious—in fact, may generally be neglected as has been shown. But the forces set up by heavy moving parts when in motion—especially in rapid motion—are quite different both in magnitude and effect. To overcome the inertia of a heavy body and start it into motion a considerable force is required; to accelerate the motion still more force is required. Such a body, then, when in rapid motion has much energy stored in it, all of which must be abstracted in order to bring it to rest; if the process of bringing to rest is accomplished quickly the force developed will be large and produce disastrous results, unless special provision is made for otherwise using it up. This force varies with the mass of the body and its acceleration; hence in a quick-running engine the inertia forces set up in the moving parts are much more serious matters than in a slow-moving engine, not only on account of the stresses set up in the parts themselves and in those adjacent to them, but because of the vibratory effects on the structure of the engine generally, as also on the structure on which it is carried. Inertia forces can be modified by reducing the weight of a moving part, but the velocity is practically fixed, being that required for the full working of the engine. Very little scope is permitted, therefore, to the designer to modify the motion of any one part.

Dealing first with those parts which have simple lineal motion, such as the piston, piston-rod, and crosshead, the valves and valve-rods, &c., the general effect can be appreciated by tracing each phase of movement. For example, at the commencement of the stroke when steam is admitted to the cylinder, it exerts a pressure on the piston and cover alike, hence the thrust on the piston (and from it to the bearings) and the resistance of the engine framing are equal but opposite in direction, and, consequently, exactly balance one another since there is no motion. There is, therefore, no load on the parts external to the engine. To produce motion, the inertia of the piston, &c., has to be overcome; if this is done by the pressure of the steam, the whole force of the steam on the piston is not transmitted through the rods to the crank, for a considerable portion of it is used up in moving the piston, &c.,* and is thus stored in the piston and its parts as energy; the load on the cover being, therefore, greater than the part transmitted, it tends to make the engine move in the reverse direction to that of the pistons. Further, the movement of the pistons is accelerated in its course by accessions of force till about the half-stroke; hence during all this time the engine is tending to move; in fact, if it is a vertical engine and the piston is on the downstroke, it will jump from its bed unless its weight is greater than the accelerating force, and if there are no holding bolts restraining it. During the latter part of the stroke the momentum of the piston, &c., tends to keep it in motion, and consequently a downward thrust on the rods is created in excess of the load on it and the cover, so that the engine now would move the other way, unless restricted by the bed on which it rests. The same phenomena occur in horizontal engines, and are of the utmost consequence in the locomotive engine; it was, in fact, with this class of engine that the effects of inertia forces were first impressed on engineers by the disastrous results with some outside cylinder horizontal engines when attempts were made to drive them at higher speeds. Fortunately the practical minds of those days soon found a remedy—one that, from its simplicity and efficiency, is still used in locomotive engines, in spite of what science has done to reveal its faults.

The general effect of the variation of momentum of the piston, &c., is to reduce the load on the crank pin at the early part of the stroke, and to increase it at the latter, with the reactive tendency to move the whole engine up and down alternately; in other words, to reduce the ratio of maximum to mean load, and cause vibration on the bed supporting the engine; as also to reduce the maximum stresses on the rods and their bolts. The vibration set up in the engine bed is, of course, communicated to the ship.

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* In fast-running engines, such as are used for generating electricity (in which the revolutions are constant while the load varies), the inertia of the pistons and rods is so great that on light loads the mere steam pressure on them would not overcome it; in fact, the piston of one engine is started and accelerated by the power transmitted from the others or from the flywheel, even when the load is considerable, hence it is most important that such engines should have good "stern-going" piston-rod guides. If an engine is moved by external means, such as a belt from another engine, the effect of inertia of the moving parts is exactly the same as if it were working under steam.

The general effect on the crank of the variation of momentum of the piston-rods, &c., will be dealt with later on, but in the meanwhile can be shown graphically by calculating the accelerating forces applied to the piston at a series of points at and from the commencement of the stroke, and taking as ordinates the equivalent pressures per square inch on the piston; H J K (fig. 198) is such a curve obtained by making E H represent the pressure on the piston required to overcome its inertia, and so also with the other ordinates. If, now, a new base line, L M, fig. 199, is taken and a curve, Ne'S P, drawn by taking as ordinates the distances intercepted by the curves H J K and Gee''P, a true representation of the variation in real push and pull of the rods is shown. The ordinates measure the combined effect of steam pressure on the piston, gravity, and inertia of moving parts. The cushioning is shown by the shaded portions at each end, S M P and R N L. In this case it may be mentioned that the diagram, fig. 197, is that of a three-crank triple-compound engine with the high-pressure crank leading, and to show the difference in effect when the medium-pressure crank leads the high pressure, the dot and dash line (fig. 199) is the curve of equivalent loads for a similar engine under that condition.

The general result of the movement of the slide valves, link motions, &c., is similar, and although their weight is less than that of the piston, &c., and their motion much more restricted, their varying momentum can set up a manifest vibration in a quick-running engine.

The connecting-rod has a complex motion, inasmuch as one end moves in a straight line while the other moves in a circle; its centre of gravity consequently moves in an ellipse whose major axis is equal to the stroke of the piston. Its motion, therefore, may be viewed as partly linear with the piston-rod, &c., and partly circular with the crank. But when in motion it also oscillates like a pendulum, so that its effect on the crosshead will be the same as that of a pendulum on its point of support; hence it will set up a horizontal force there in addition to the downward one due to gravity. The force will vary with the mass and the horizontal acceleration; it will set up a horizontal vibration in the engine and increase the stresses on the rod due to cross bending; hence in very fast-running engines the connecting-rod should be made as light as possible consistent with strength. A short rod is, of course, lighter than a longer one, but it gives a greater thrust, due to steam pressure, and, consequently, tends to increase vibration transversely.

The crank, including arms and pin, moves in a circle, and consequently centrifugal forces are set up which tend to press the shaft against the journals in the radial direction of the crank pin for the time being. The weight of the crank pin brasses and that part of the connecting-rod surrounding them moves in a circle, and causes further centrifugal forces to act on the crank-shaft. These parts by their momentum also set up tangential forces, which, with the resisting force at the bearings, form a couple tending to keep the shaft in motion; but as the direction changes, so the reacting force changes in direction, and this also tends to rack the bearings and foundation, as well as to set up vibration.

Balancing.—From the above causes a reciprocating engine of any kind when in motion sets up vibrating forces which are only extinguished by doing work of some kind, mostly, however, of a mischievous character. Even the slow-running paddle engine developed the vice, which, when the engine is nearly horizontal and has a single cylinder, as in certain Clyde steamers, is very perceptible fore and aft ways. When the engine was directly coupled to the screw shafting, and its revolutions consequently three or four times that of the older engines, vibration and its effects were soon noticed, and although some general idea of the causes was grasped, and attempts were made to grapple with the evil by fitting balance weights opposite the cranks or in the turning wheel at the end of the crank shaft, no true insight seems to have been gained; consequently, for more than half a century engineers generally were content to put down the vibration of screw steamers as due to the propeller, and to admit that the paddle wheel was in that respect the superior one for passenger steamers.

Now, any force which has to be extinguished by doing useless work is a force wasted, and if the work done is injurious to engine or hull, the loss is worse still. For this reason the engineers of the past are to blame for not having found means to prevent such waste, and still more so, those who now neglect the efficient methods which have been devised for preventing such forces coming into existence, and for rendering them harmless when developed.

Mr. Arthur Rigg called attention to the subject in his able work so far back as 1878, but it is owing to Mr. Yarrow's genius for investigation that the question was put in practical form, and to Mr. Otto Schlick's analytical ability that methods were devised whereby balancing was no longer an interesting workshop experiment, but an exact science and a fine art; it has been rendered even more so by the later contributions of Prof. Dalby and others to the *Transactions of the Institution of Naval Architects, &c.*

To understand clearly what is involved in the balancing of an engine, it will be best to take, first, the common case of a double-acting single-crank engine. Suppose it to have a flywheel with a weight near its rim of such a size and position that at half-stroke it statically balances the crank, rods, and piston. When this engine is in motion the momentum of the principal parts is roughly balanced for turning by that of the balance weight; but a couple is formed by joining the centre of the gravity of the weight by a line going diagonally across the engine to its centre line. The shaft and its bearings will, therefore, be racked by this cross-section couple, and the engine tend to tilt and twist on its bed. Moreover, since the momentum of the balance weight is constant tangentially, while that of the pistons, rods, &c., ceases at the ends of the stroke, there must be an excess momentum of the balance weight at those periods which, in the vertical engine, tend to make it move horizontally, and in the horizontal engine vertically. This evil is made apparent in a locomotive engine by what is called "rail hammering," from the blows the rail gets from the wheels with the balance weights when the engine passes the dead centres.

If the balance weight is placed on one crank-arm only, as was sometimes done for cheapness, similar actions were set up, but less in degree. Two balance weights, one on each arm, make the proper balance, but still fail to avoid the horizontal overbalance when the crank passes the dead points and the consequent horizontal vibration. If only balance weights opposite the cranks can be provided, a compromise must be arrived at, so

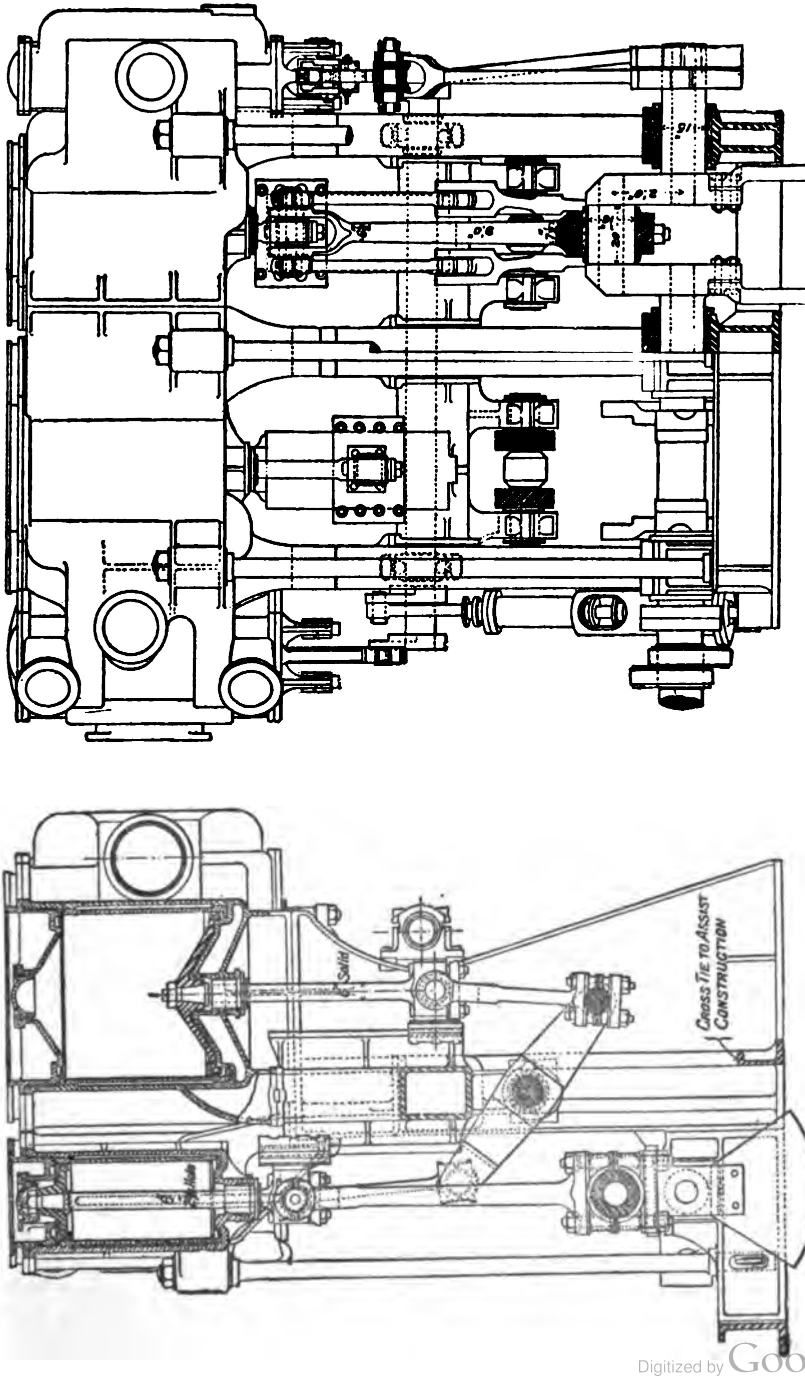


Fig. 200.—MacAlpine's Quadruple Balanced Engine, Cylinders 27", 39", 56", and 80" diameter × 48" stroke.

that their weight is such that the horizontal vibration due to overbalance is reduced to that of the vertical vibration caused by underbalance. If, however, the engine has an air pump, &c., worked by means of levers so that they move in the opposite direction to that of the piston, their weight may be so adjusted that their momentum balances that of the piston and rods, while the balance weights on the crank balance the crank, pin brasses, and connecting-rod end. There will, however, still remain a couple tending to rock the engine athwart ship.

Mr. Macalpine has designed a quadruple engine (fig. 200) with the cylinders in pairs, one behind the other, and each pair operating on the same crank by means of rocking levers like those of the air pump. This gives, no doubt, a very well balanced arrangement fairly free from vibration, but the ratio of maximum to mean torque is greater than that of a three-crank engine.

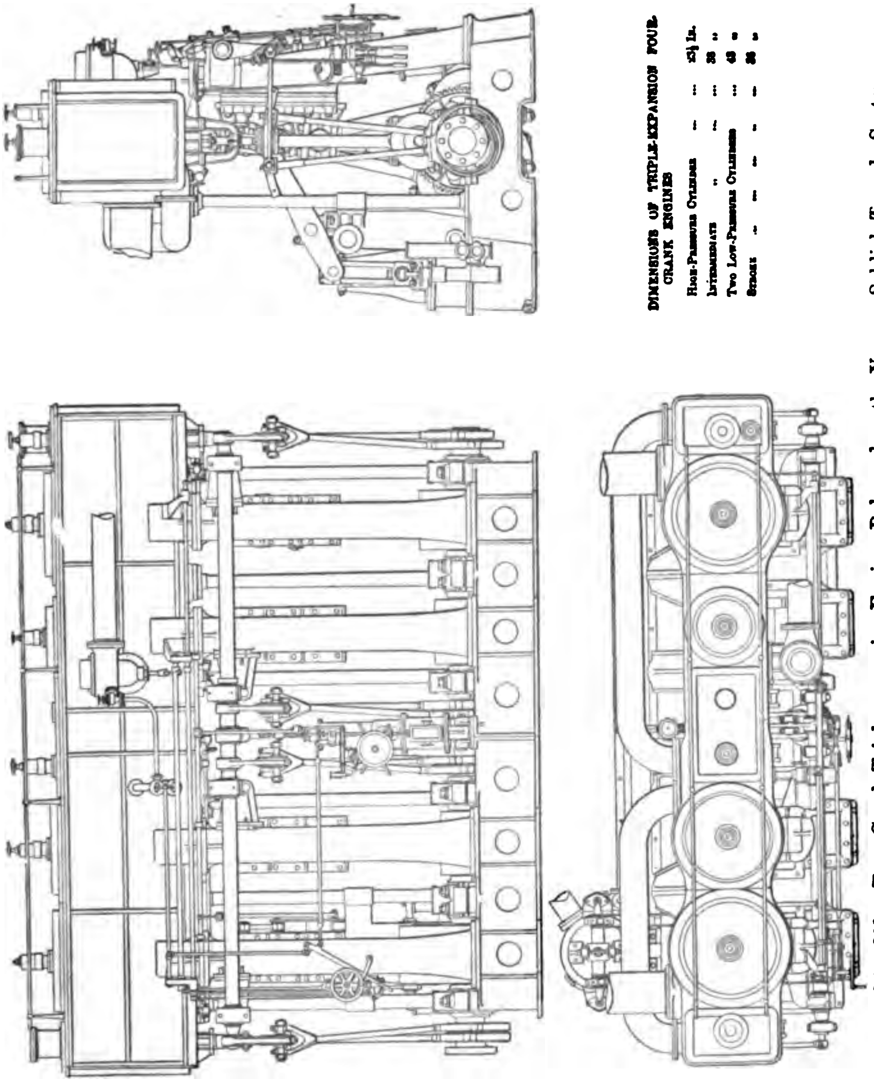
Further, the motion of the valve and gear in this single engine will create inertia forces which will tend to set up vertical vibration, and should be taken account of especially if it is a fast-running one. This can be done by finding the resultant forces from all the parts, and placing the balance weights accordingly.

Take now the case of a two-crank engine. If the cranks are opposite one another they will balance for rotation and so require no weights, and if the moving parts are of equal weight their relative momenta will also only tend to make the engine rock in a fore and aft direction on its foundation and not jump on it; it is true, however, that this tendency to rock will be prevented by the holding down bolts, but with the vibration of the ship as a consequence. Such vibration, however, will not be so troublesome as is the case with an unbalanced single engine tending to jump. In this instance the balancing might be effected by a weight in the flywheel at a slight expense, or else by some form of "bob" weight, as suggested by Mr. Yarrow, placed beyond one of the cranks so as by its mass and velocity to balance the momentum on the other crank-pin. This, of course, leads naturally to a three-crank engine with the two outer ones *opposite* the middle crank and the weight of the moving parts of the middle engine equal to those of the two outer engines together, these being of equal weight.

This form of engine would seem to be perfectly balanced, and it would actually be so if it had slot crossheads instead of connecting-rods; as it is, the acceleration of the piston of the middle engine at the commencement of its downstroke is greater than that of the wing engine pistons at commencement of their upstrokes with a consequent loss of balance and vibration.

If a two-crank engine has its cranks at right angles it will be, from the ordinary point of view, a better balanced engine than one with the cranks opposite, although there may be more vibration set up by it. The fact is, that the torque is much more even in this case—that is, the ratio of maximum to mean twisting moment is much less; and the engine runs more uniformly without a flywheel. This little fact enforces the necessity of clearly distinguishing between the balancing for even circular motion and the balancing necessary to prevent vibration either vertical or horizontal, although generally the balancing to prevent vibration tends to improve the even running of the engine. Balancing of these engines may be done by means of a pair of weights, one at each crank, in such a way that the vertical is not extinguished at the expense of excessive horizontal vibration.

The same process must be gone through with a three-crank engine whose cranks are at an angle of 120° with one another—viz., to ascertain the kinetic resultants of the system, and balance them wholly or partially by means of balance weights placed on the first and third cranks of such a size and position that the engine will neither rock fore and aftways nor jump on its seat; and, further, it must not have excessive horizontal



DIMENSIONS OF TRIPLE-EXPANSION FOUR-CRANK ENGINES

High-Pressure Cylinder	23½ in.
Intermediate "	38 "
Two Low-Pressure Cylinders ...	43 "
Stroke	36 "

Fig. 201.—Four-Crank Triple-expansion Engines Balanced on the Yarrow-Schlick-Tweedy System.

vibration. The methods of arriving at these results are set out fully later on.

A four-crank engine, however, is the easier to deal with, as in that case contiguous engines can have their cranks opposite one another so as to balance vertically and nearly balance horizontally and have the minimum rocking moment. Also, as the extreme cranks are at right angles they give a smaller rocking couple than if the angle were larger. Mr. John Tweedy devised an arrangement of cylinders and cranks in a four-crank triple engine whereby a good balance is obtained without the use of any balance weights or other addition to the engines. In his design the low-pressure cylinders with their gear, as being the lightest, are placed at the ends with the high-pressure and medium-pressure cylinders between them. The first and second cranks are nearly, but not quite, opposite one another, as are also the third and fourth; the second is not quite at right angles with the third; in fact, the cranks are all so set as to give the best balance of the whole. This is as perfect an arrangement as can be obtained without the use of balance weights and is such as to be practically without vibration.

Fig. 201 shows such an engine and fig. 201*a* the arrangement of the cranks to suit the conditions of that particular engine. This design is now a favourite one with all requiring a steady running engine with the least vibration of ship, and is being fitted generally into passenger steamers, where the comfort of the passengers and the preservation of the paint work and decoration, to say nothing of the structure of the ship, is of prime importance.

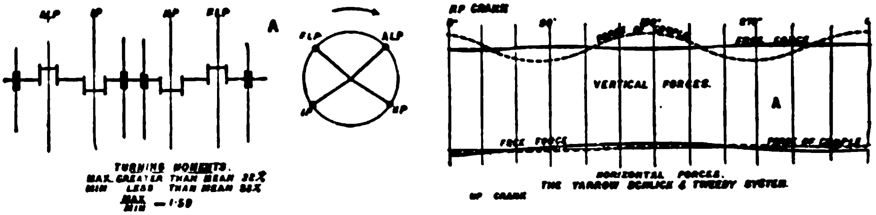


Fig. 201*a*.—Position of Cranks and Curves of Force, &c., of Triple-expansion Engine Balanced on the Yarrow-Schlick-Tweedy System.

In warships, where accuracy of fire may mean victory, well balanced engines must be fitted. The Admiralty therefore insist on all engines being as perfectly balanced as possible.

As, however, a four-crank engine is generally more costly and occupies more space than a three-crank one and the ratio of its manifold torque (or twisting moment) to the mean torque is greater than that of a three-crank engine, the latter will continue to be largely used. The following investigations and methods of balancing such an engine will be both of interest and importance.

Figs. 202 and 203 show an engine designed and patented by M. E. Wigzell, whereby a nearly perfect balance is obtained by having one cylinder above another with the pistons moving in opposite directions. Here again, the obliquity of the connecting-rods prevents absolutely perfect balance; in practice, however, the engine behaves as if it were perfectly balanced. This engine has the further merit of occupying very little fore and aft space and is readily started from any position.

1. **Preliminary Definitions.**—The direct object of what is termed "balancing an engine," is to prevent the effect of the motion of its moving parts being felt by surrounding objects, and not to eliminate any particular stresses in parts of the engine itself. Neither has the object any connection with fluctuation of speed, the theory and action of flywheels not being related to the subject.

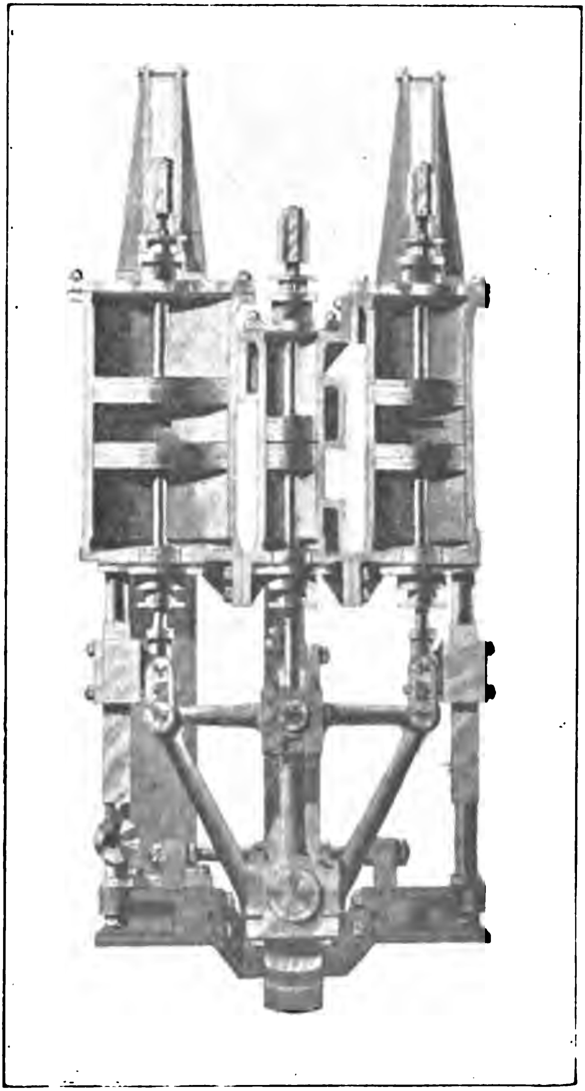


Fig. 202.—Wigzell's Engine. Transverse Section.

It is a natural *consequence* of balancing an engine that certain stresses are considerably reduced; that others remain as they were, and that some possibly even may be increased.

Another point, which may not at first sight appear evident, is that an engine may be very much out of balance in itself, that is, if placed on a rigid foundation, it causes very severe stresses on the holding-down bolts, yet if placed in certain positions on a flexible foundation may appear to be in a perfectly balanced condition; that is to say, it will run without producing any vibrations of the foundation, and in fact can run safely without holding-down bolts.

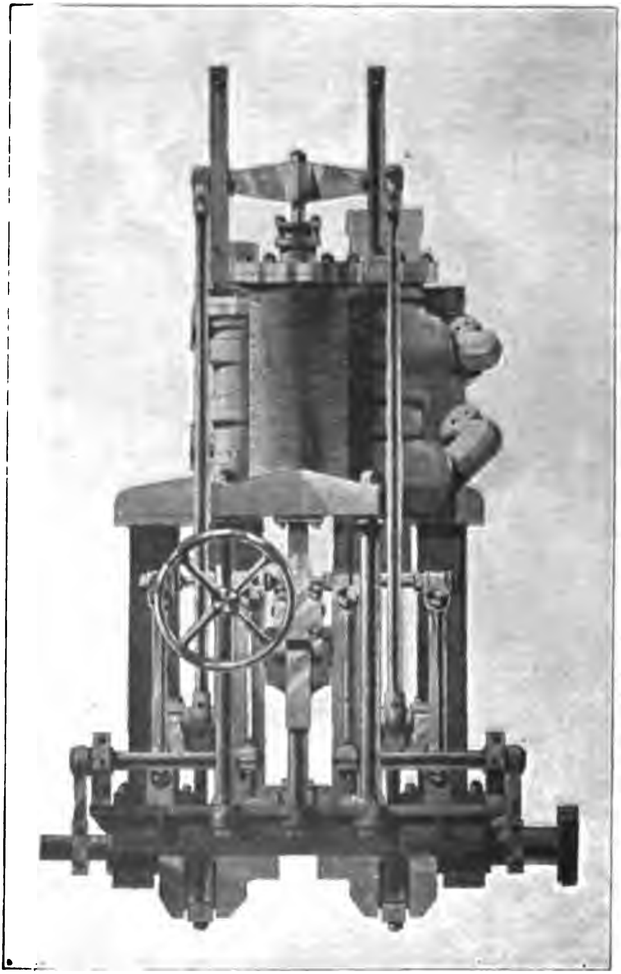


Fig. 203.—Wigzell's Engine. Longitudinal Elevation.

The problem of balancing an engine need not necessarily be dependent on this phenomenon, and very fortunately so, because, although a ship is certainly a flexible foundation for an engine to rest on, so little of the true nature of its flexibility is subject to calculation that, for new ships particularly, any calculations involving this property are really mere guesswork and therefore better left alone.

If, however, the engine be perfectly balanced in itself it can cause no vibration to the foundation, whatever may be the nature of its flexibility.

In practice it is impossible to perfectly balance an ordinary engine and it will be shown here within what limits and to what extent an engine can be balanced. All problems will relate particularly to the vertical engine, but the principles involved apply equally well to horizontal and inclined engines. Before proceeding with this, however, attention must be drawn to the fact that the problem of balancing an engine is not in the least concerned either with steam loads and twisting moments due to the same or with the speed of the engine, the dynamical equilibrium of the moving parts being quite independent of how fast or how slow or how varying the speed of the engine may be. Comparative values of the speeds of the various parts are certainly required, for the magnitude of any resultant forces and couples are determined by the speed, but unless those absolute values are required, only comparative values of the speeds of the parts are essential.

As many engineers, some even of high standing, are apt at times to confuse the matter, and imagine that there is some mysterious connection between steam pressures and balance weights, an enlargement of what has been said under "Momentum," Section 2, showing that there is no connection between the two, is given in the following simple demonstration.

Let fig. 204 represent a diagrammatic engine. Let the steam load on the piston be + P. Then the load on the top cover is - P, and, confining attention to the vertical forces, the vertical pressure at the main bearings is + P. That is, the steam load causes no pressure of the engine on the foundation, A.

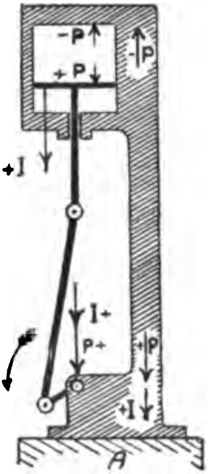


Fig. 204.

The moving parts of the engine, however, are not weightless, also they never have a uniform velocity in any given direction (in this case, the vertical)—that is, they always have a certain acceleration, positive or negative. Now, since

$$\text{force} = \text{mass} \times \text{acceleration}$$

another vertical force is consequently introduced, and is—

$$I = \frac{\text{weight of moving parts} \times \text{vertical acceleration}}{g}.$$

Suppose that when the parts are in the position shown, this force equals + I lbs. acting downwards. Since the whole of the moving parts are concerned, this is equivalent to a load + I pressing the piston down as shown. Now it is self-evident that there cannot be any force - I acting on the cylinder cover, therefore this force + I remains alone and its effect is consequently to make the engine heavier by + I lbs. At other points the inertia force will act upwards instead of downwards, and it will also have a continuously varying value.

There are, therefore, up and down forces due to the *weight and velocity of the moving parts only*, acting on the engine as a whole and recurring every revolution.

As these are the forces that tend to make the foundation vibrate, the balancing of an engine consists of counteracting the forces so that the moving parts are really made as if weightless when considered dynamically in certain given directions.

It has also been stated above that the problem of balancing has no direct relation to the speed of the engine. This will be obvious on consideration, for any counteracting weights that may be applied must necessarily have their velocity imparted to them by the engine itself, and the factor of velocity is consequently eliminated from both sides of the representative equations.

2. **Harmonic Motion.**—The theorem of harmonic motion is stated here because of its great importance and the direct bearing it has on the present subject.

Suppose a point P (fig. 205) to move uniformly round the circle $X\,Y\,X^1$. Draw two rectangular axes O X, O Y, and P N perpendicular to O Y. Then if P has a uniform angular velocity N will move along $Y\,O\,Y^1$, with a motion that is called "Simple Harmonic Motion."

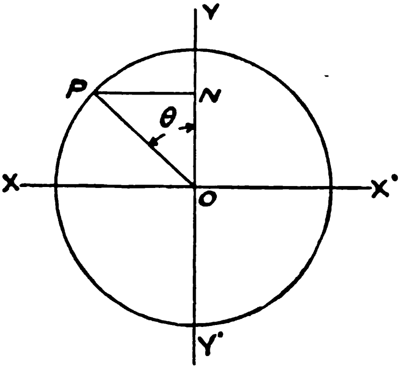


Fig. 205.

Let $OP = r, \qquad POY = \theta,$

then $ON = r \cos \theta.$

Let $v =$ linear velocity of P,

then Velocity of $N = -v \sin \theta$

and Acceleration of $N = -\frac{v^2}{r} \cos \theta.$

The proof of these will be found in most treatises on theoretical mechanics.

P is called the *generating point*.
The angle θ is called the *phase*.
r is called the *amplitude*.
The time of one revolution of P is called the *period*.

A curve of the displacement of N from O or of the acceleration of N may be drawn if the abscissæ be made proportional to θ and the ordinates proportional to the displacement or acceleration. Such a curve is shown in fig. 206 by $x \cos \theta$.

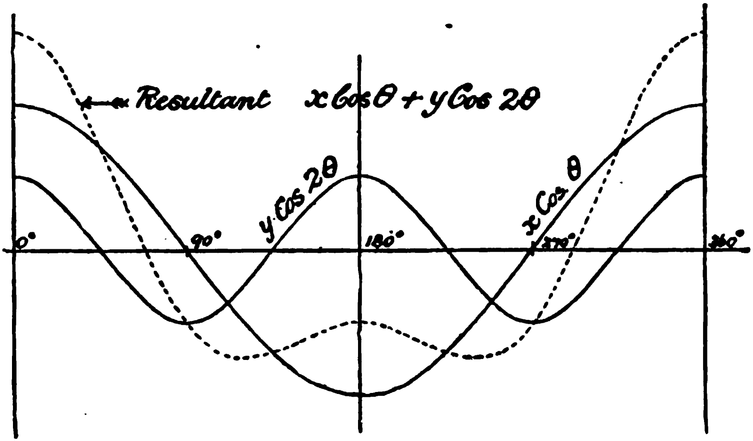


Fig. 206.

3. To Combine Two or more Harmonic Motions of the same Period—

Let P and Q (fig. 207) be the generating points of two harmonic motions along the axis O Y.

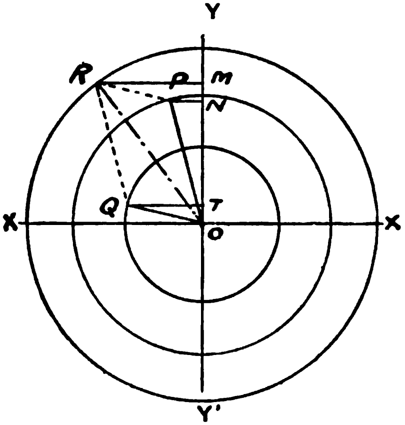


Fig. 207.

Then, since the angular velocities of O P and O Q are the same and are of the same sign, the angle P O Q is constant.

Complete the parallelogram P O Q R. The linear velocities of P and Q are proportional to O P, O Q respectively, and are perpendicular to them.

Therefore O R is proportional to their resultant which is perpendicular to it.

Because O P, O Q and angle P O Q are constant, O R is constant and has the same angular velocity as P and Q. Therefore R is the generating point of the resultant harmonic motion.

The combination of any number of harmonic motions of the same period and sign is analogous to the combination of any number of coplanar forces by the polygon of forces.

4. The Inertia of the Various Parts of an Engine will now be dealt with separately.

Inertia of Piston, Rod, and Crosshead.—These fittings all being connected together rigidly, and having only one line of motion, the velocity and acceleration of the whole is the same at all points.

(1) If their movement is controlled by a connecting-rod of infinite length they have a simple harmonic motion, and the acceleration at any angular position θ of the crank from the top centre is therefore

$$f = -\frac{v^2}{r}\cos\theta \text{ (par. 2).} \quad - \quad - \quad - \quad - \qquad (1)$$

(2) If their movement is controlled by a connecting-rod of definite length the simple harmonic motion is at once destroyed and the expression for the acceleration is a very complicated one, so much so that it is impossible in practice to use it in its complete form for either algebraical or graphical calculations.

The expression correctly stated in definite form is—

$$f = -\frac{v^2}{r}\left(\cos\theta + \frac{rl^2\cos 2\theta + r^3\sin^4\theta}{(l^2 - r^2\sin^2\theta)^{\frac{3}{2}}}\right) \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \qquad (2)$$

Or, written in the form of a series,

$$\begin{aligned} f = -\frac{v^2}{r}\Big\{ & \cos\theta + \cos 2\theta\left(\frac{1}{s} + \frac{1}{4s^3} + \frac{15}{128}\cdot\frac{1}{s^5} + \ldots\right) \\ & + \cos 4\theta\left(\frac{1}{4}\cdot\frac{1}{s^3} + \frac{3}{16}\cdot\frac{1}{s^5} + \ldots\right) \\ & + \&c. \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \Big\} \quad \cdot \quad \cdot \qquad (3) \end{aligned}$$

Where l = length of connecting-rod
and s = length of connecting-rod ÷ radius of crank.

As neither of these expressions are convenient to use, it is necessary to adopt the nearest approximation that will admit of a tangible interpretation.

The first approximation is $f = -\frac{v^2}{r}\cos\theta$, which is the same as (1), and is therefore the value of f to use when the connecting-rod is of infinite length, or, in other words, when its "obliquity" is neglected.

The second approximation is

$$f = -\frac{v^2}{r}\left(\cos\theta + \frac{\cos 2\theta}{s}\right) \quad - \quad - \quad - \qquad (4)$$

and this is sufficiently accurate for all ordinary calculations and makes an almost complete allowance for the obliquity of the connecting-rod, for the other terms of the series diminish very rapidly in value. This expression (4) will therefore be used in the problems of finding the actual values of the disturbing forces in an engine.

The error involved by using this expression instead of the complete series or equation (2) is, if $s = 4$, 1·56 per cent., and if $s = 5$, 0·8 per cent. This error is therefore very small indeed, and in any graphical calculations would hardly be appreciable.

Means of Balancing the Piston and Rod.—Expression (4) may be written in the form $f = -\frac{v^2}{r}\cos\theta - \frac{v^2}{rs}\cos 2\theta$. Cos θ and cos 2θ are both simple harmonic functions but with this difference, cos θ has *one* period per revolution of the engine and cos 2θ has *two* periods per revolution. This will be seen at a glance in fig. 206, where both these curves and their resultant are drawn.

Let M_1 = weight of piston, rod, and crosshead. Now, in order to balance the inertia force, $F = -\frac{M_1}{g}\cdot\frac{v^2}{r}\cos\theta - \frac{M_1}{g}\cdot\frac{v^2}{rs}\cos 2\theta$, two weights must be fitted so that one of them has an acceleration in the same line of motion as the piston, equal to $+\frac{v^2}{r}\cos\theta$, and the other has an acceleration also in the same line of motion, equal to $+\frac{v^2}{rs}\cos 2\theta$, both weights being equal to M_1.

Or, the weights W_1 and W_2 may have different values and radii of influence (r_1 and r_2), so long as

$$-\frac{M_1 v^2}{g r}\cos\theta = \frac{W_1 v_1^2}{g r_1}\cos(180^\circ + \theta),\ i.e.,\ W_1 r_1 = M_1 r$$

and

$$-\frac{M_1 v^2}{g s r}\cos 2\theta = -\frac{W_1 v_2^2}{g s r_2}\cos 2(180^\circ + \theta),\ i.e.,\ W_2 r_2 = M_1 r.$$

Considering the first force, it will be obvious that if a weight W_1 be fitted to revolve with the crank-shaft and opposite to the crank, it will have the required vertical acceleration and, therefore, the required counteracting inertia force.

This is the method that is in the majority of cases adopted.

It has one drawback, however; another inertia force is introduced in a horizontal direction due to its harmonic motion in that direction.

This force is equal to $\frac{W_1 v_1^2}{g r_1}\sin\theta$.

The main result of this method is really to overbalance the engine in a horizontal direction, as will be seen in a subsequent paragraph.

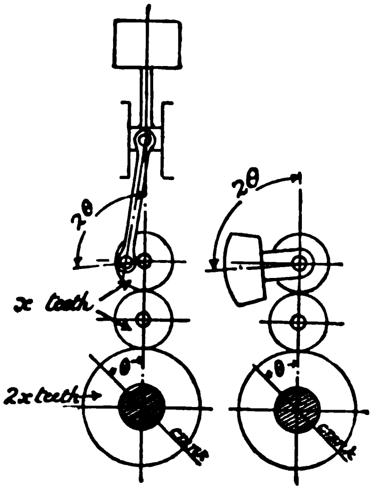
Practically, two weights would have to be fitted each equal to half W_1, and on each crank web.

A more correct method is that suggested by Mr. Yarrow to fit two (for practical convenience) "bob" weights, driven by eccentrics, keyed on the shaft. $\frac{W_1}{2}$ should equal the weight of the bob, eccentric, and rods, and the eccentrics of throw r_1 must be placed opposite to the crank. The only error involved in this method is the slight horizontal inertia force due to the weight of the eccentrics and obliquity of the eccentric-rods.

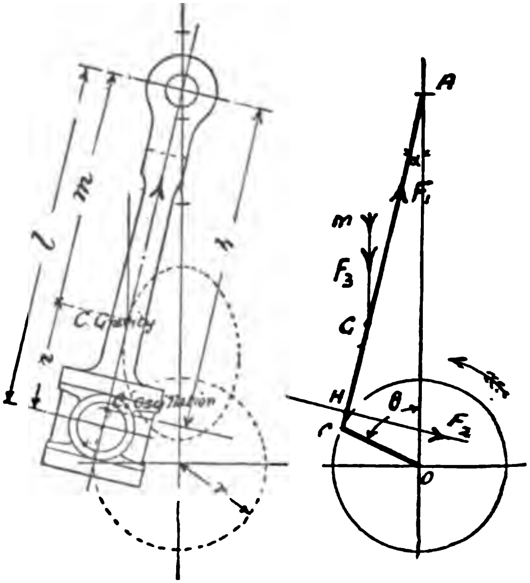
Considering the second force $-\frac{M_1 v^2}{g s r}\cos 2\theta$, a mechanical difficulty is confronted here in producing a fitting that shall move with an acceleration proportional to cos 2 (180° + θ). Two ways in which this could be done are shown in figs. 208, 208a. The figures are self explanatory. Another way would be by means of a cam fixed to the shaft that would cause a weight to move up and down twice per revolution and satisfy the condition that its acceleration would be always proportional to cos 2 (180° + θ).

There are possibly other mechanical devices for attaining the same object.

Now, these methods, although very well in theory, would be altogether too inconvenient for actual practice. Thus it is that this part of the disturbing force on an engine is neglected as far as actual balancing is concerned. Doing so is generally termed "neglecting the obliquity of the connecting-rod."



Figs. 208 and 208a.



Figs. 209 and 209a.

5. Inertia of the Connecting-rod.—In finding the acceleration of the connecting-rod a problem of extreme complexity is presented, and it is usual and necessary, therefore, in practice to make certain approximations and assumptions in treating the subject. The connecting-rod having its top end moving in a straight line and its bottom end in a circle, has consequently a different total acceleration at all points in its length. Adopting the approximation in par. 4, equation 4, the top end has an acceleration the same as the piston-rod, which is therefore $f = -\frac{v^2}{r}\left(\cos\theta + \frac{\cos 2\theta}{s}\right)$.

The bottom end moving with a uniform angular velocity about the axis of the shaft has a vertical acceleration $-\frac{v^2}{r}\cos\theta$ and a horizontal acceleration $\frac{v^2}{r}\sin\theta$.

Any point on the connecting-rod between the centres will have an elliptic path, the minor axis of which varies from $2r$ to O, and the vertical acceleration varies from $-\frac{v^2}{r}\cos\theta$ to $-\frac{v^2}{r}\left(\cos\theta+\frac{\cos 2\theta}{s}\right)$. If the mass of the whole rod be supposed concentrated at the centre of gravity the acceleration of the whole rod vertically will be

$$-\frac{v^2}{r}\left(\cos\theta+\frac{n}{l}\cdot\frac{\cos 2\theta}{s}\right). \qquad \text{(fig. 209)} \qquad (5)$$

If it is desired to find the true value of the inertia of the rod in order to combine it with the "steam" twisting moments, or for reasons of other interest, the strict investigation is outlined in the footnote, but as far as balancing is concerned, it is out of the question for practical reasons to use such elaboration.

Considering expression (5) for the acceleration, it is obvious from the remarks on the inertia of the piston-rod (§ 4) that a balance weight can be fitted to the engine that shall have an acceleration proportional to cos θ, but that it is impossible practically to fit one having an acceleration proportional to cos 2 θ. The part $-\frac{v^2}{r}\cdot\frac{n}{l}\,\frac{\cos 2\theta}{s}$ has then perforce to be neglected.*

---

* Let θ be angular position of the crank from the top centre and α the angle the connecting-rod makes with the centre line. Then at any instant the angular velocity of the connecting-rod is $\frac{d\alpha}{dt}$ and the angular acceleration of the connecting-rod is $\frac{d^2\alpha}{dt^2}$. There are three forces on the rod which determine its resultant acceleration (fig. 209*a*):—

$F_1=\frac{Mm}{g}\left(\frac{d\alpha}{dt}\right)^2$ acting along the rod.

$F_2=\frac{Mm}{g}\,\frac{d^2\alpha}{dt^2}$ acting at right angles to the rod through H.

$F_3=\frac{M}{g}f$ acting through G parallel to the centre line.

Where M = weight of the rod.
f = acceleration of rod vertically.
g = acceleration of piston-rod.

It may easily be shown that—

$$\frac{d\alpha}{dt}=\frac{v\cos\theta}{\sqrt{l^2-r^2\sin^2\theta}}$$

$$\frac{d^2\alpha}{dt^2}=-\frac{v^2(l^2-r^2)\sin\theta}{r(l^2-r^2\sin^2\theta)^{\frac{3}{2}}}$$

$$f=-\frac{v^2}{r}\left(\cos\theta+\frac{rl^2\cos 2\theta+r^3\sin^4\theta}{(l^2-r^2\sin^2\theta)^{\frac{3}{2}}}\right)$$

By substitution, each of the three forces can be found. In addition to the forces there is the deadweight of the rod = M acting vertically through G. To find the effect on the twisting moment, the moments of the four forces must be taken about O for different values of θ and added algebraically to the remainder of the twisting moment diagram. The proof of the above formulæ may be found in advanced treatises on mechanics and steam.

Means of Balancing the Connecting-rod.—Neglecting, therefore, the part involving cos 2θ, the vertical acceleration of the rod is $\frac{v^2}{r}\cos\theta$. To balance by a rotating weight, a weight must be placed in the same position as that for balancing the piston, &c. If M_3 equal the weight of connecting-rod, then the weight W_3 is given by $W_3 r = M_3 r$. This weight will partly balance the rod in a horizontal direction. It will, however, really overbalance horizontally, because, although the whole rod has a vertical acceleration $-\frac{v^2}{r}\cos\theta$, only part of it has an acceleration $\frac{v^2}{r}\sin\theta$ (which has the same maximum value or amplitude as $\frac{v^2}{r}\cos\theta$, its phase being 90° behind). Therefore, if it is desired to balance any particular engine horizontally in preference to vertically, the full weight W_3 must not be put on, but only a portion of it, which for a fairly accurate approximation may be such that $\frac{m}{l} M_3 r = W_3 r$. (fig. 209).

This is one of the assumptions alluded to at the beginning of the paragraph, and really amounts to dividing the connecting-rod into two parts, and assuming the part $\frac{n}{l} M_3$ to be concentrated at the crosshead and $\frac{m}{l} M_3$ to be concentrated at the crank-pin.

Professor Dalby gives a division of the rod that is said more accurately to represent the case, the division being as follows:—At the crank pin $M_3\left(\frac{m \times h}{l^2}\right)$. The remainder at the crosshead, h, being the distance from crosshead centre to centre of oscillation.

This necessitates an experiment with one of the actual rods to find the centre of oscillation.

The finding of the centre of gravity does not require any special experiment, for in ordinary shop transit the slings are usually placed in this position. In case that either observations are not readily available, the following table of typical connecting rods is given :—

(v. Fig. 213.)	Connecting-Rod to		
	Fig. 52.	Fig. 58.	Fig. 214.
Position of centre of gravity from top centre, .	·675 l	·58 l	·643 l
,, ,, oscillation ,, ,, .	·88 l	·912 l	·99 l
$\frac{m \times h}{l^2}$, ...	·594	·528	·637

If it is desired to balance the engine as far as possible, both vertically and horizontally, then the reciprocating part of the connecting-rod must be balanced by means of reciprocating bob weights as explained and the rotating part must be balanced by a rotating weight.

6. Inertia of Crank.—The balancing of the crank is the simplest part of the whole problem. The unbalanced part, of course, consists of the crank webs beyond the shaft and the crank-pin. The acceleration of the centre of gravity vertically is $f = -\frac{v^2}{r}\cos\theta$, and horizontally $f = \frac{v^2}{r}\sin\theta$, or, if referred to a horizontal axis, the horizontal acceleration is $f = -\frac{v^2}{r}\cos\theta$. Thus to balance both vertically and horizontally a weight must be placed opposite the crank-pin acting at radius r_1 such that its weight is given by $W_4 r_1 = M_4 R$, where R is the distance of the centre of gravity of the crank webs, &c., from the shaft centre.

7. Inertia of Valve Gear.—The simple valve gear is, of course, a slider-crank chain like the piston, connecting-rod, and crank, and its treatment is therefore precisely the same.

Practically it is not convenient to fit special balance weights for the valve gear, and they are generally embodied in those for the main parts of engine. As this method introduces a disturbing couple on the engine which must be eliminated, the problem of combination will be treated at length in par. 9.

Again, most marine engines are fitted with Stephenson's link motion. It is impossible to use the formula for the acceleration of the valve in any method of balancing; in fact, the solution of the displacement, velocity, and acceleration of the slide valve is an extraordinarily complicated operation, especially if slot links are fitted.

It is usual, therefore, to adopt the following assumptions in dealing with this part of the engine:—

1. *Reciprocating Masses.*
Ahead, . . Valve.
Spindle.
Half the link.
Link block.
0·5 eccentric-rod if notch up gear; or
0·4 eccentric-rod if direct gear.
Half the bridle-rods = 1 bridle-rod if connected to "ahead" end.
Astern, . . Half the link.
0·5 eccentric-rod if notch up gear; or
0·4 eccentric-rod if direct gear.
1 bridle-rod if connected to "astern" end.

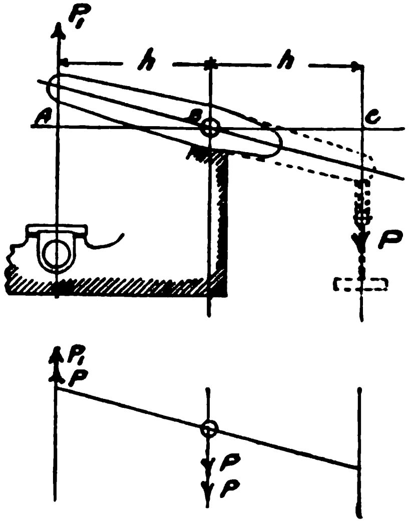
2. *Rotating Masses.*
Ahead, . . Remainder of eccentric-rod.
Eccentric sheave.
Eccentric strap.
Astern, . . Remainder of eccentric-rod.
Eccentric sheave.
Eccentric strap.

8. Inertia of Air Pump.—The air pump when out of centre with the engines, and having an opposite motion to that of the particular engine that drives it, presents a rather peculiar problem when its effect on the engine support or foundation is considered conjointly with the means of balancing.

Let W_5 W_6 be the equivalent weights of parts when having an acceleration f equal to that of the piston-rod (fig. 210). Then A B = B C.

Let the inertia forces due to W_5, W_6 be P, P_1 respectively $\left(P = W_5 \frac{f}{g}\right.$ and $\left.P_1 = W_6 \frac{f}{g}\right)$, and suppose them to act as shown in fig. 210 when the lever is in that position.

The force P is equivalent to a transverse couple 2 P*h* on the engine and a downward force P acting through the rocking shaft bearing (fig. 210*a*). There is thus a total force $P + P_1$ acting upward with the piston-rod inertia force and a force 2 P acting downward through the bearing.



Figs. 210 and 210*a*.

Means of Balancing.—There is no practicable method of balancing the transverse couple, except perhaps by fitting an equal air pump on the opposite side of the engine and driven from the same crosshead. This would, however, be an extremely awkward fitting.

Now, if *h* is an appreciable dimension compared with the breadth of the ship, the effect will be to produce torsional vibrations of the ship; but if *h* be very small compared with the same, there is on practically the same centre line a downward force 2 P and an upward force $P + P_1$. The difference of these forces is the force causing vibrations of the same kind as those produced by the engine. In practice it will be found that W_5 and W_6 are very nearly equal to one another, and in that case the two inertia forces neutralise each other, and the air pump may consequently be neglected.

In the greater number of fast-running engines *h* will, however, be of appreciable magnitude, and it is, therefore, usual to make the addition $W_5 + W_6$ to the engine reciprocating weights.

Note.—When in the remainder of this chapter the term "perfectly balanced" is used without comment, it may be taken as meaning that the parts under consideration are perfectly balanced as far as the limits just discussed enable them to be.

9. Single-crank Engine and Valve Gear balanced by Rotating Weights only.—Reduce all weights of moving parts to equivalent weights driven by cranks of the same radius, say the crank radius *r*.

Let M_1 = weight of engine reciprocating parts at radius *r*.
M_2 = „ „ rotating „ „
M_3 = „ valve reciprocating „ „
M_4 = „ „ rotating „ „

Imagine a plane O X Y (fig. 211) through *any* point O on the crank shaft and at right angles to it. This plane will be called the reference plane.

Fig. 211.

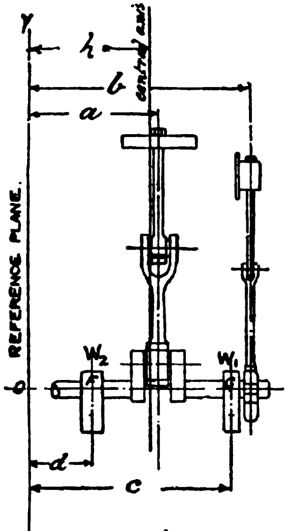


Fig. 211a.

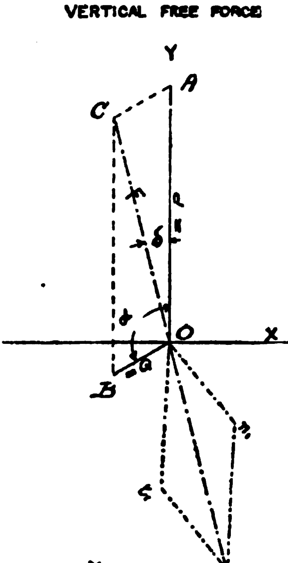
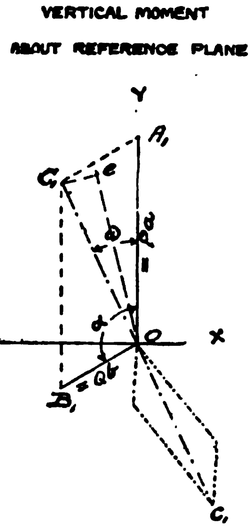
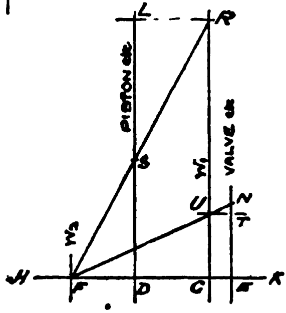
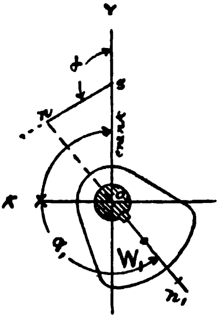


Fig. 211b.







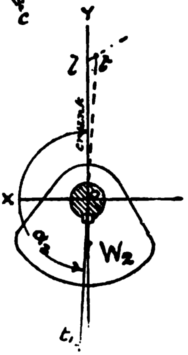


Fig. 211c. Fig. 211d. Fig. 211e.

Case 1.—Suppose it is desired to balance the engine as completely as possible in a *vertical* direction in preference to the horizontal direction.

By the theory of couples any number of coplanar forces can be reduced to a single force acting at any point accompanied by a couple.

In the engine under consideration there is at any time a free vertical force $-\frac{v^2}{g\,r}(M_1+M_2)\cos\theta - \frac{v^2}{g\,r}(M_3+M_4)\cos(\theta+\alpha)$, and a couple referred to the reference plane

$$= -\frac{v^2}{g\,r}(M_1+M_2)\,a\cos\theta - \frac{v^2}{g\,r}(M_3+M_4)\,b\cos(\theta+\alpha)$$

As the force and couple cannot be balanced by a single force, because they are not in the same phase, two weights at least must be employed to balance the engine. A couple as well as a force can be represented by a straight line of length proportional to its moment. It is usual in theoretical mechanics for this line, which is called the axis of the couple, to be drawn anywhere at right angles to the plane of the couple, but in order not to confuse the subject it will be found much more convenient to draw this line in the plane of the couple.

Project the lines of cranks on to the reference plane (shown in front elevation on figs. 211*a* and 211*b*), and measure off parts O A and O B equal to $(M_1 + M_2)$ and $(M_3 + M_4)$ respectively. Measure off OA_1 and OB_1, equal to $(M_1 + M_2)\,a$ and $(M_3 + M_4)\,b$ respectively. Each of these quantities multiplied by $-\frac{v^2}{gr}$, of course, gives the maximum inertia forces and couples due to them, and the angle α is also the phase of those forces and couples of the crank behind those of the valve gear. $-\frac{v^2}{gr}$ being common to all, balance weights included, need not therefore enter into any calculations. Then A, B, A_1 and B_1, are the generating points of simple harmonic motions along O X, O Y. Compound these by the problem in par. 3. Their resultants are C and C_1 respectively.

Suppose it is convenient to place the rotating balance weights at points C_1 and F on the shaft. Then, stated definitely, the problem is to find the values of the weights W_1 W_2, and the angles at which they are to be fixed relatively to the crank such that the resultant of the vertical free force produced by their rotation is equal and opposite to O C, and that the resultant of their moments about the reference plane is equal and opposite to OC_1.

For convenience let $M_1 + M_2 = P$, and $M_3 + M_4 = Q$. Let the angles W_1 W_2 make with the crank be α_1 α_2 respectively.

Then these values may be found by solving the equations—

$$P + Q\cos\alpha = -(W_1\cos\alpha_1 + W_2\cos\alpha_2) \quad - \quad - \quad - \quad - \quad - \quad (1)$$

$$Pa + Qb\cos\alpha = -(W_1 c\cos\alpha_1 + W_2 d\cos\alpha_2) \quad - \quad - \quad - \quad - \quad - \quad (2)$$

$$W_1^2(c-d)^2 = P^2(a-d)^2 + Q^2(b-d)^2 - 2PQ(a-d)(b-d)\cos\alpha \quad (3)$$

$$W_2^2(c-d)^2 = P^2(c-a)^2 + Q^2(b-c)^2 - 2PQ(c-a)(b-c)\cos\alpha \quad (4)$$

Care must be taken that the distances are positive or negative according to their sense. These are very easily solved if the known values of the constants be substituted immediately.

The *graphical solution*, however, is more quickly arrived at.

Draw a base line H K, and a pair of axes O X, O Y (figs. 211*c*, 211*d*, and 211*e*). On H K mark off the engine centres, &c., to scale, and erect perpendiculars as shown.

On either of the balance weight centres (say that of W_1) mark off G R = P, and G U = Q.

Draw horizontals through R and U. Join R F, cutting D L in S. Join U F, cutting E T in N. Then, by the principle of the "lever," the part of D L or P that must be apportioned to W_1 is D S, and the part to W_2 is S L.

Similarly, the part of E T to be apportioned to W_1 is E N and the part to W_2 is T N. E N is necessarily greater than E T, because the valve centre is outside both the centre of W_1 and W_2.

To find W_1 mark off (fig. 211*d*) os = DS and sn = E N. Join no and produce. Then $on = W_1$ and on_1, its direction relatively to the crank line oy.

To find W_2 mark off ol = S L (fig. 211*e*), and lt = − T N. T N being negative the direction must be in the opposite direction to sn.

Then $W_2 = ot$ and ot is its position relatively to the crank line O Y.

If the work has been done correctly, the resultant of the two weights W_1 W_2 will be found to be equal and opposite to O C, and the resultant of their moments about the arbitrary reference plane will be found to be equal and opposite O C_1 (figs. 211*a* and 211*b*).

The centre of gravity of W_1 and W_2 must be at radius r, but, if necessary for it to be at radius r_1, then $W_1r = w_1r_1$, and $W_2r = w_2r_2$. Thus the engine is balanced perfectly in a vertical direction for both the distributing couple and the free up and down force.

Case 2.—Suppose it is desired to balance the engine as completely as possible in a *horizontal* direction in preference to the vertical direction.

Here the rotating parts of the engine have simply to be dealt with, because the reciprocating parts have no resolved part horizontally.

Apply the same process as for Case 1, but instead of using $M_1 + M_2$ and $M_3 + M_4$ use M_2 and M_4 respectively.

It will now be seen, as intimated in the latter part of par. 5, that the engine is over-balanced horizontally when completely balanced vertically by rotating balance weights.

Case 3.—Suppose it is desired to balance the engine as completely as possible in both directions, the balance weights being rotary only.

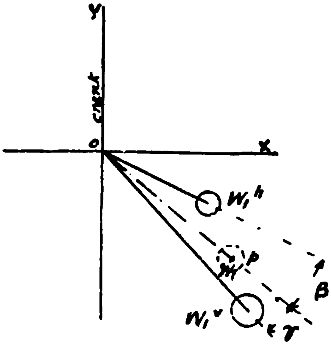


Fig. 212.

The only thing that can be done here is to effect a compromise.

Find, as in Cases 1 and 2, the weights necessary for balancing in both directions separately. Let these weights be W_1^v and W_2^v for the vertical balance and W_1^h and W_2^h for the horizontal balance. Draw their centre lines as in fig. 212. They will most probably not be on the same centre line. Divide the angle between them so that $\frac{\gamma}{\beta} = \frac{W_1^h}{W_1^v + W_1^h}$. Then Y$op$ is the best angle for the weight to be fitted. Also make $op = \frac{W_1^v + W_1^h}{2}$. Then op is the value of the weight W_1 to be fitted.

Treat W_2^v and W_2^h in a similar manner.

The engine is thus incompletely, and at the same time as completely, balanced as possible under the condition of having only rotating balance-weights.

10. Single-crank Engine and Valve Gear Balanced by Bob Weights and Rotating Weights combined.—The engine may be balanced perfectly both vertically and horizontally by the above combination.

In order to do so the bob weights are made to balance the reciprocating parts, and the rotating weights are made to balance the rotary parts.

To find the bob weights proceed exactly as before, using M_1 and M_2 for P and Q. The weights found are those which would be driven by eccentrics of throw r. This radius will in most cases be too large for convenience. If r_1 be the convenient throw, then $W_1 r = B_1 r_1$, and $W_2 r = B_2 r_1$, B_1 and B_2 being the new weights which should include the weight of the eccentric and rod. To find the rotating balance weights again proceed as before, using M_2 and M_4 for P and Q respectively, but also taking into account the weight of the bob weight eccentrics, as they introduce a horizontal disturbance which must be eliminated.

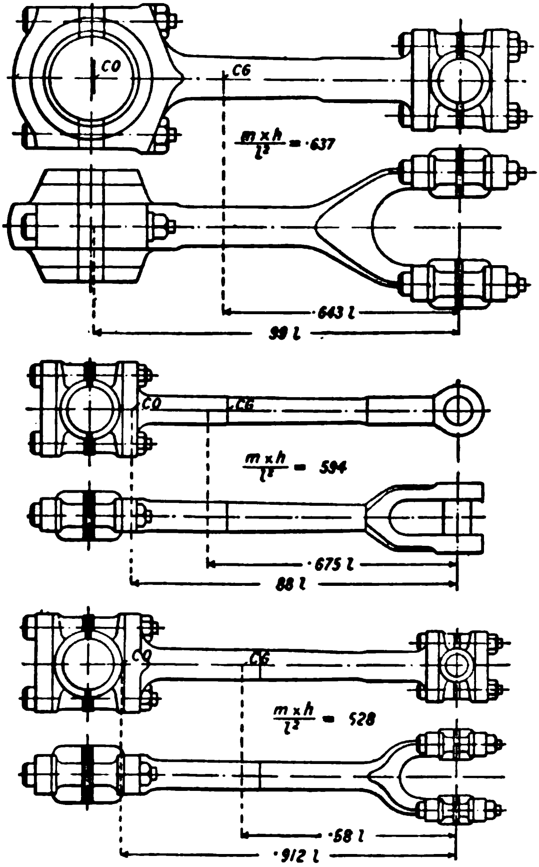

CO
C6
$\frac{m \times h}{l^2} = \cdot 637$
·643 l
·99 l
CO
C6
$\frac{m \times h}{l^2} = \cdot 594$
·675 l
·88 l
CO
C6
$\frac{m \times h}{l^2} = \cdot 528$
·58 l
·912 l

Fig. 213.

The solution of the problem of balancing by this particular method, of course, depends on the fact that the bob weights have no resolved part horizontally.

11. To Balance an Engine of any number of Cranks.—As the method of solving this problem is the same in principle as has just been described in pars. 9 and 10, the case will be most easily explained by means of an example.

In the example the engine will be balanced both vertically and horizontally, and the combined system of bob and rotating weights must, therefore, be used. Should, however, the designer of any particular engine decide that he cannot very well adopt the most complete method, he will understand from the previous paragraphs what sacrifices will have to be made if only one kind of balancing device is used. The methods of construction are precisely the same, although more extensive.

The example (fig. 214) is a three-cylinder engine with ordinary link motion for each valve. Two bob weights will be fitted, one at each end of the engine, and four balance weights, one on each crank web of the outside engines. It will also be decided to balance half the rotating masses by the H P and L P forward weights, and half by the H P and L P after weights.

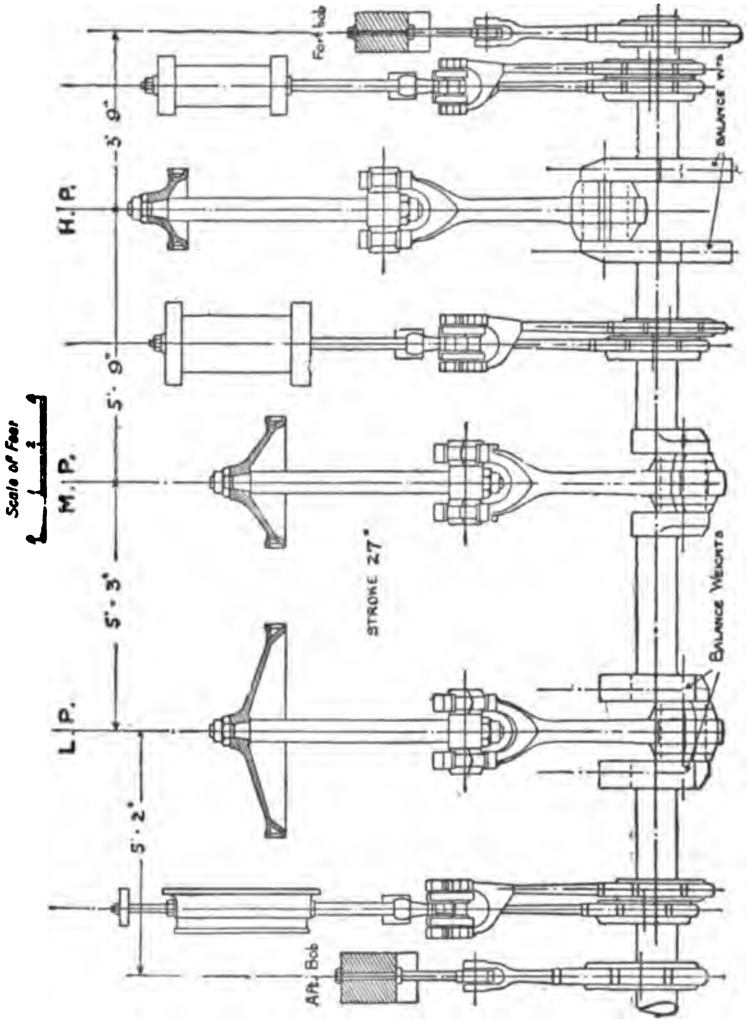


Fig. 214.

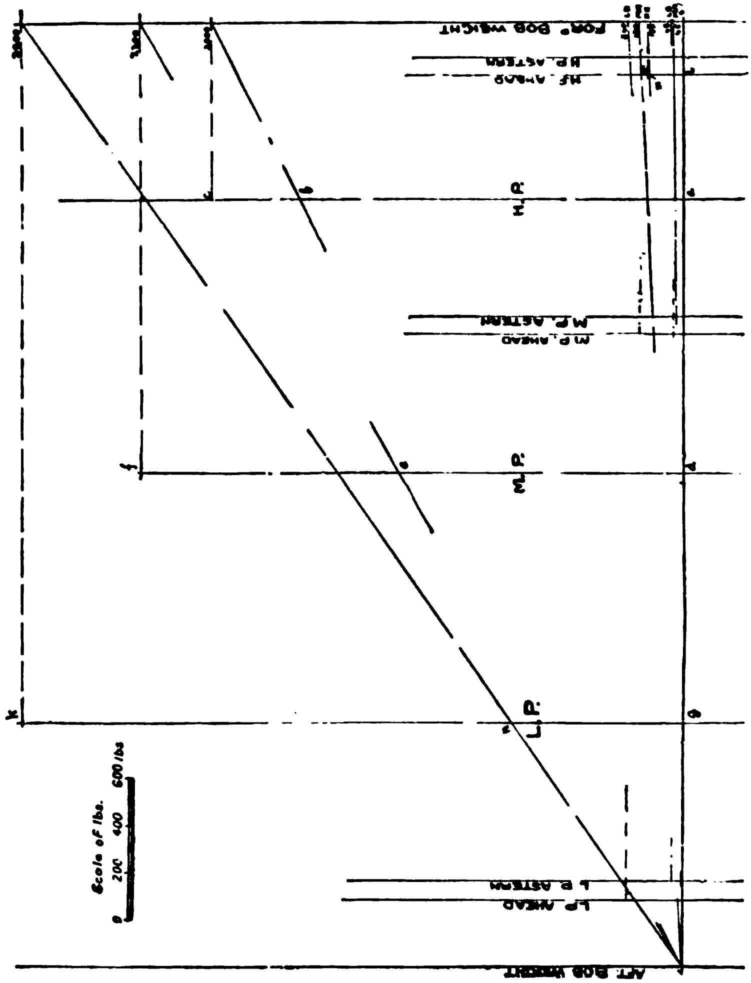

FOR? BOB WEIGHT
H.P. ASTERN
H.P. AHEAD
H.P.
M.P. ASTERN
M.P. AHEAD
M.P.
L.P.
L.P. ASTERN
L.P. AHEAD
AFT BOB WEIGHT
Scale of lbs.
0
200
400
600 lbs
k
f
c
b
a
d
g

Fig. 215.

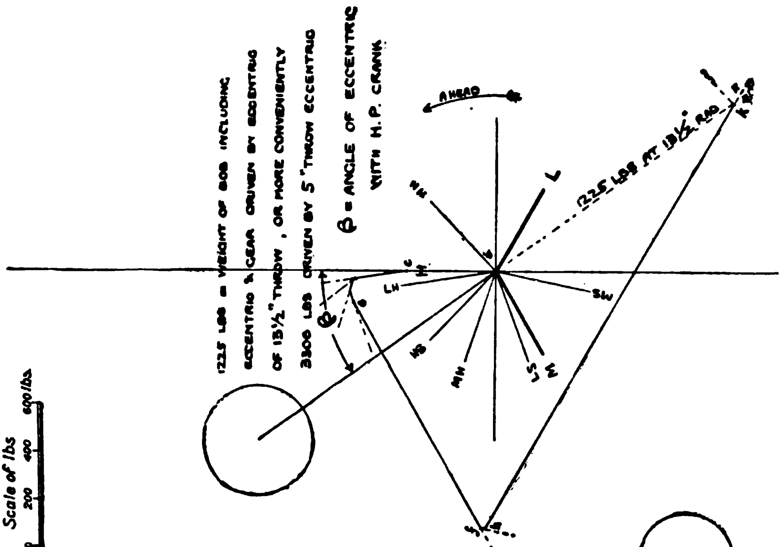

Scale of lbs
0
200
400
600 lbs
1225 LBS = WEIGHT OF BOB INCLUDING ECCENTRIC & GEAR DRIVEN BY ECCENTRIC OF 13½" THROW, OR MORE CONVENIENTLY 3300 LBS DRIVEN BY 5" THROW ECCENTRIC
β = ANGLE OF ECCENTRIC WITH H.P. CRANK
AHEAD
1225 LBS AT 13½" RAD
L
MH
H
LH
SW
W
LS
MH
HS
β

Fig. 215b.

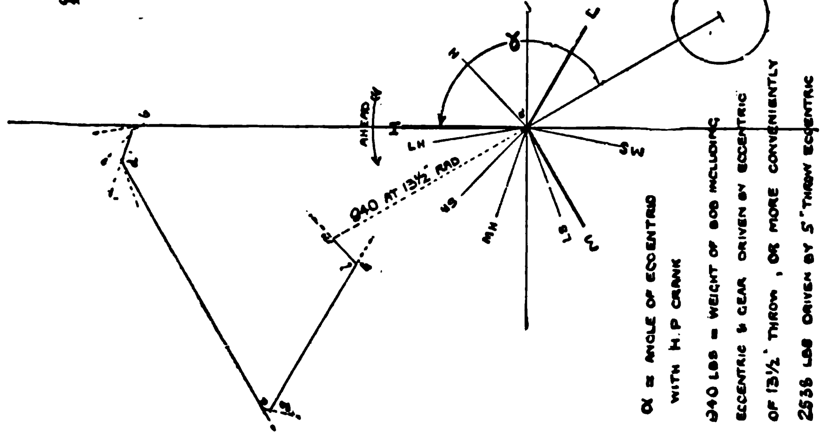

α = ANGLE OF ECCENTRIC WITH H.P CRANK
940 LBS = WEIGHT OF BOB INCLUDING ECCENTRIC & GEAR DRIVEN BY ECCENTRIC OF 13½" THROW, OR MORE CONVENIENTLY 2538 LBS DRIVEN BY 5" THROW ECCENTRIC
AHEAD
940 AT 13½" RAD
α
H
L
H
LH
SW
HS
MH
LS
W

Fig. 215a.

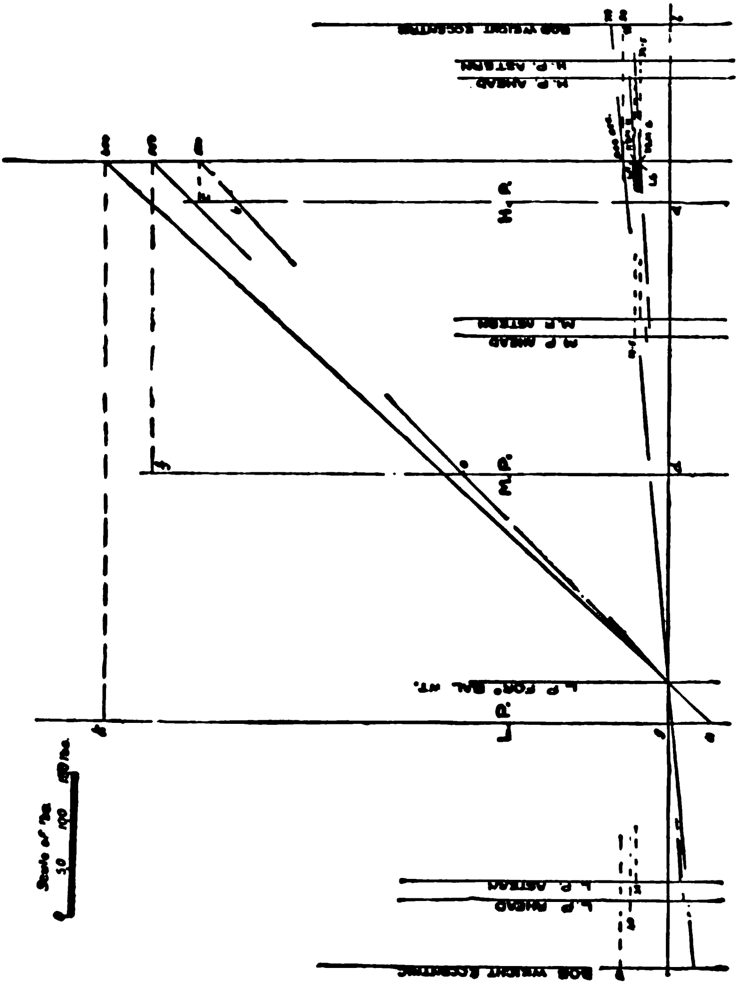

BOB WEIGHT ECCENTRIC
H.P. ASTERN
H.P. AHEAD
H.P.
M.P. ASTERN
M.P. AHEAD
M.P.
L.P. FOR'D. EN. HT.
L.P.
L.P. ASTERN
L.P. AHEAD
BOB WEIGHT ECCENTRIC
Scale of lbs
0
50
100
150 lbs.

Fig. 216.

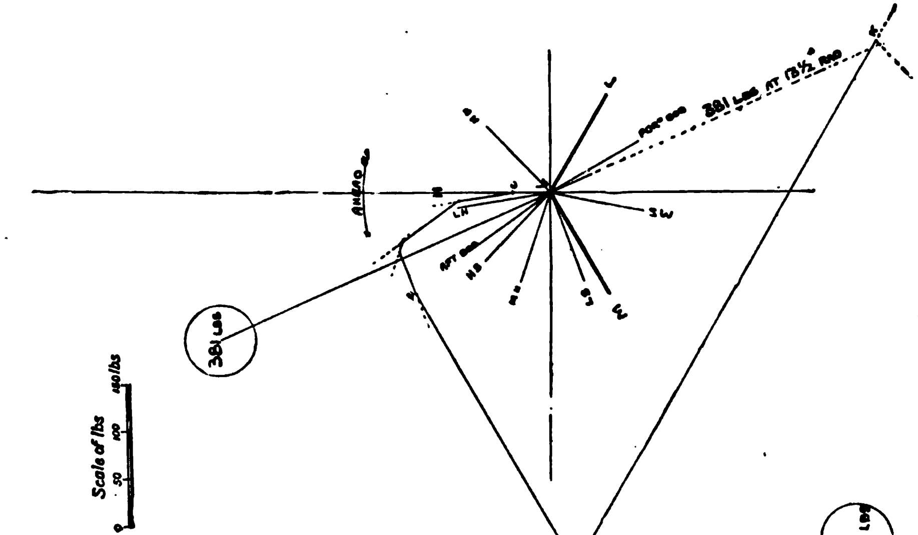


Fig. 216b.

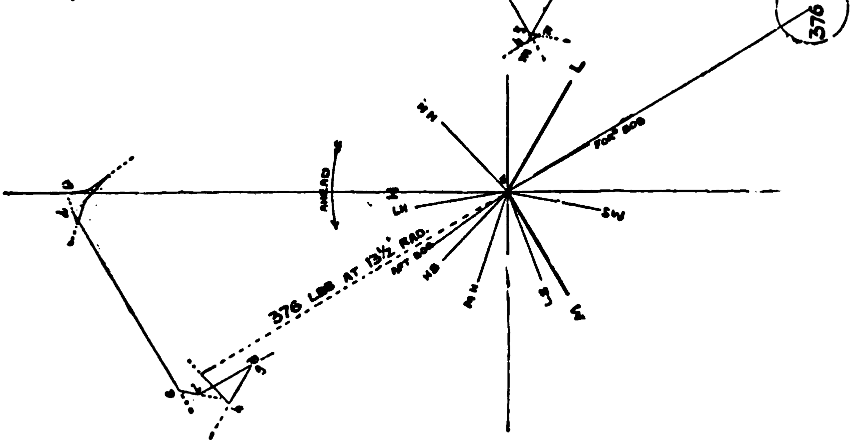


Fig. 216a.

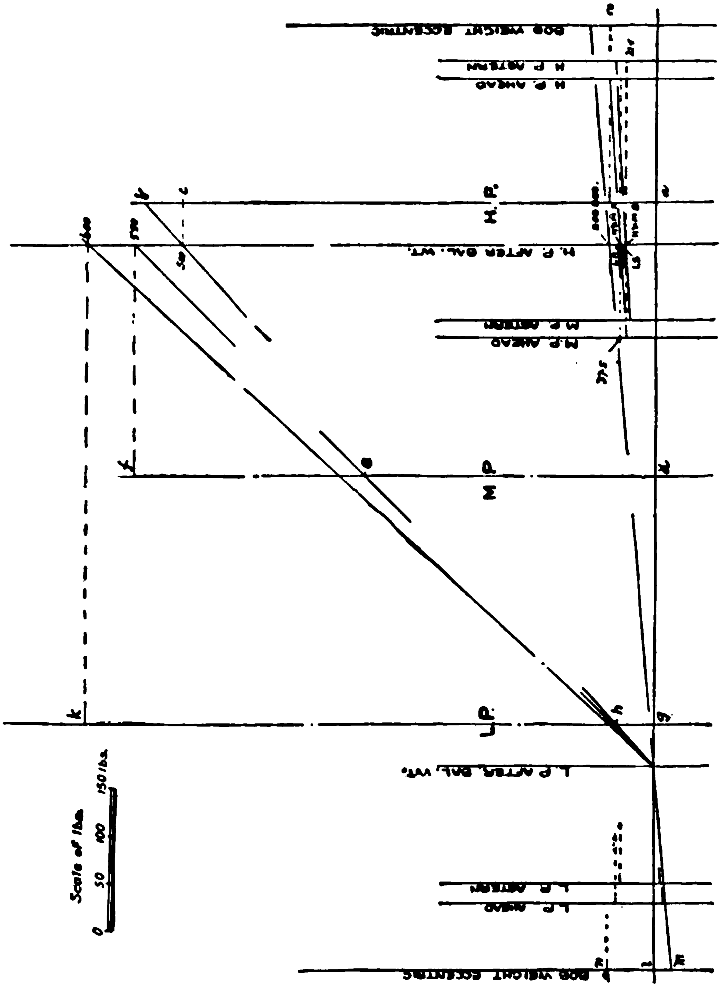

BOB WEIGHT ECCENTRIC
H.P. ASTERN
H.P. AHEAD
H.P.
H.P. AFTER BAL. WT.
M.P. ASTERN
M.P. AHEAD
M.P.
L.P.
L.P. AFTER BAL. WT.
L.P. ASTERN
L.P. AHEAD
BOB WEIGHT ECCENTRIC
Scale of lbs.
0
50
100
150 lbs.
a
b
c
d
e
f
g
h
k
l
m
n
600
520
500

Fig. 217.

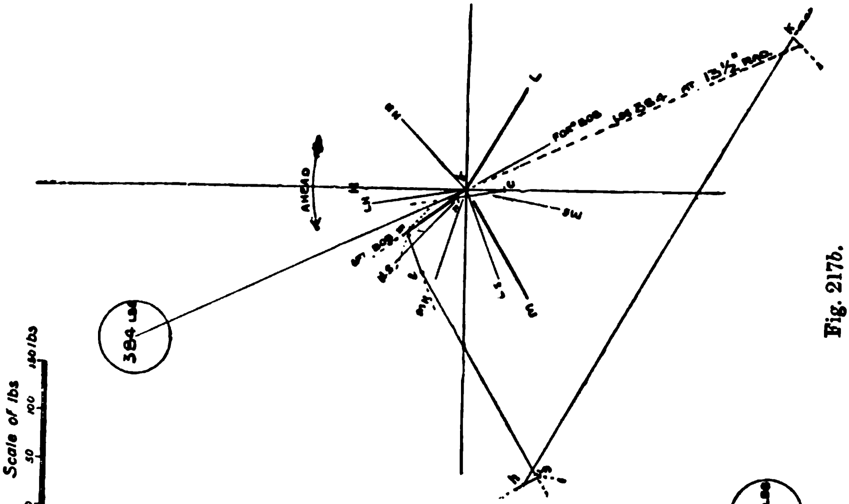


Fig. 217b.

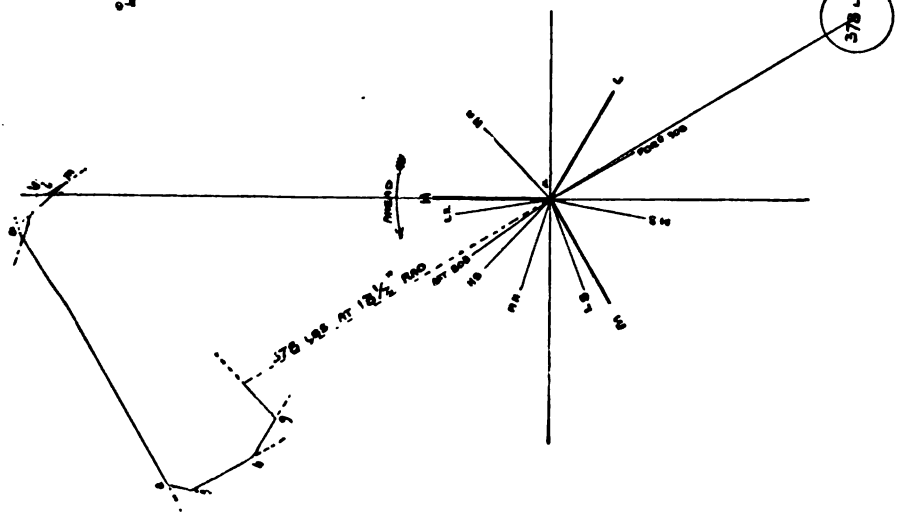


Fig. 217a.

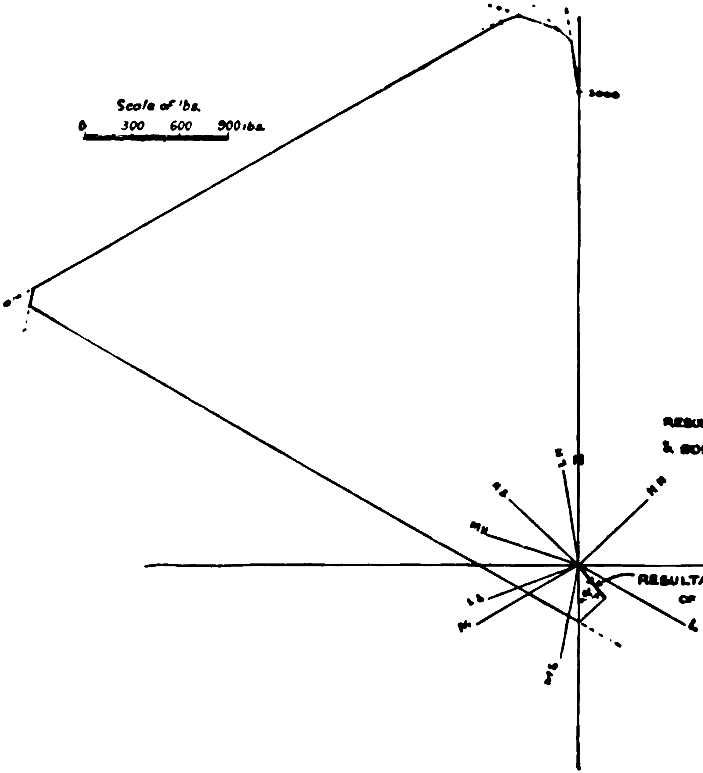


Fig. 218.

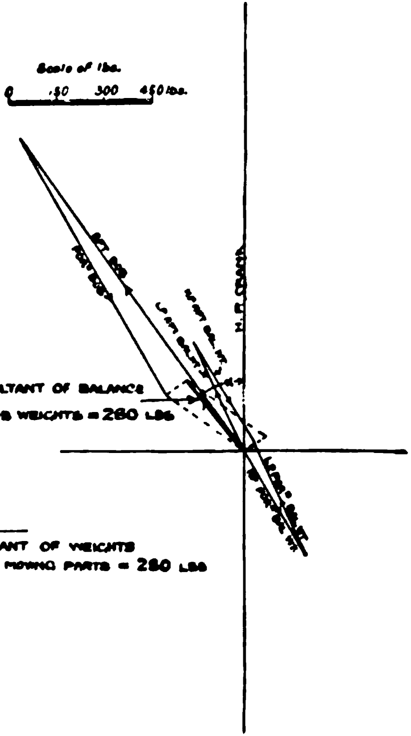


Fig. 219.

The weights of the moving parts are tabulated as follows, and the diagrams are drawn in figs. 215 to 219.

TABLE LII.—Balancing Engines: Weights of Moving Parts.

	Reciprocating Weights.			Rotating Weights.				
	Actual Weight of Reciprocating Parts.		Equivalent Weight at Radius of Crank, 13½ Inches.	Actual Weight of Rotating Parts.		Equivalent Weight at Radius of Crank, 13½ Inches.	Half Rotating Weight.	Sum of Reciprocating and Rotating Weights.
	Lbs.	At Radius, Ins.	Lbs.	Lbs.	At Radius, Ins.	Lbs.	Lbs.	Lbs.
H.P. engine, .	2,000	13½	2,000	1,000	13½	1,000	500	3,000
,, ahead, .	698	3	155	338	3	75	37½	230
,, astern, .	202	3	45	292	3	65	32½	110
M.P. engine, .	2,300	13½	2,300	1,100	13½	1,100	550	3,400
,, ahead, .	855	3	190	338	3	75	37½	265
,, astern, .	202	3	45	292	3	65	32½	110
L.P. engine, .	2,800	13½	2,800	1,200	13½	1,200	600	4,000
,, ahead, .	1,036	3⅛	240	345	3⅛	80	40	320
,, astern, .	207	3⅛	48	302	3⅛	70	35	118
Forward bob,	2,538	5	940	Eccentric.		100	50	...
After bob, . .	3,306	5	1,225	,,		100	50	...
H.P. forward,	Balance Weights.			376	13½	...	...	...
,, after, .				378	13½	...	...	..
L.P. forward,				381	13½	...	..	...
,, after, .				384	13½	...	...	...

Figs. 215, 215*a*, 215*b* are for finding the bob weights;

Figs. 216, 216*a*, 216*b* for the H.P. and L.P. forward balance weights.

Figs. 217, 217*a*, 217*b* for the H.P. and L.P. after balance weights.

Fig. 218 finds the resultant of the vertical free force by using the sum of the reciprocating and rotating parts.

Fig. 219 finds the resultant of the vertical free force of all the bob and balance weights.

If the work is done correctly these two resultants will be found to be equal and opposite to each other.

The figures and construction will be found self-explanatory after reading pars. 9 and 10.

12. So far, it has been seen that, in order to balance an engine, it is only necessary to know the relative values of the inertia forces and not the actual value. But, in order to compare one design of an engine with another, it is necessary to know the actual values (or at least relative values referred to a common basis) of the resultant disturbing forces and couples.

For instance, in comparing submitted designs of engines for any particular ship, it is obvious that, in a general sense, the less the resultant forces and couples are, the less the vibration, and, therefore, the better the engine.

Referring to par. 4, *et seq.*, dealing separately with the inertia of the various parts, and having balanced the engine as perfectly as practically possible, it will be seen that the disturbing forces left unbalanced are:—

For the piston, rod, and crosshead a vertical force $-\frac{M_1 v^2}{g r} \frac{\cos 2\theta}{s}$.

For the connecting-rod a vertical force $-\frac{M_3 v^2}{g r} \frac{n}{l} \frac{\cos 2\theta}{s}$.

For the valve gear there is a very minute quantity, which is a function of $2\,\theta$ and the higher multiples of θ, and which may very reasonably be neglected.

The residual resultants in the cases of the single cylinder engine will now be considered and the methods and principles of constructing the various resultants will apply equally well to a multi-cylinder engine.

13. *Case* 1 (corresponding to Case 1, p. 553).—The residual vertical inertia force is $x \cos 2\,\theta$ where $x = {}^*\left(M_1 + \frac{n}{l}\text{ connecting-rod}\right)\frac{v^2}{grs}$ and is constant. There is no residual couple, seeing that the very small force left from the valve gear is neglected.

Put $\theta = 0$, then $\cos 2\,\theta = 1$ and x is the amplitude, par. 2, of the force, and is the quantity that can be used for comparative purposes.

To draw the curve $x \cos 2\,\theta$, it is easier to do so graphically than to go to the labour of calculating ordinates for various values of θ. Draw a quadrant of a circle of radius x to any suitable scale (fig. 220). Divide the arc into a convenient number of parts and drop perpendiculars on to the axis O X as shown. Draw a base line A B and divide it into *twice* the number of parts the complete circle is divided into, because the curve has two periods per revolution. Mark the divisions and degrees and erect perpendiculars to AB through the division points as shown. Mark off A 1 = O 1, $b\,2 = b\,2$, $c\,3 = c\,3$, and so on. Then the curve through E, 3, F is the required curve, the ordinates of which give the residual resultant disturbing or unbalanced inertia force at any position θ of the crank from the top centre. The curve of resultant moment about any reference plane may be drawn in a similar manner—for instance, that plane through a node of the ship. As in this case the engine is overbalanced horizontally there is a residual horizontal disturbing force.

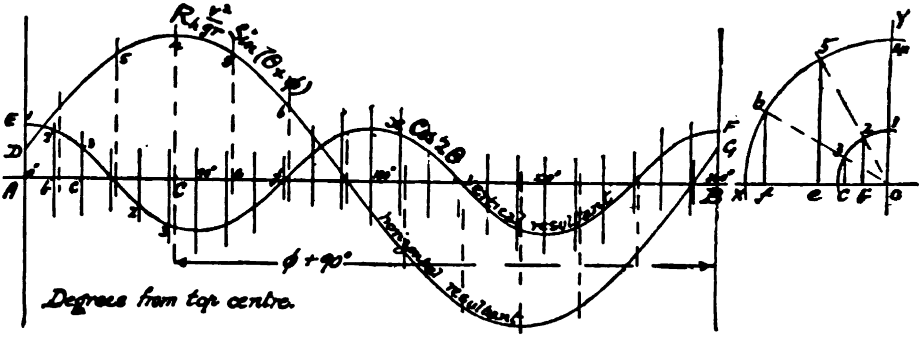


Fig. 220.

Let the resultant of the balance weights W_1 and W_2 be W_e (fig. 221). Also, let the resultant of the rotating parts of the engine be R, both these resultants being drawn in their proper relation to the crank. Find the resultant R_k of W_e and R. It makes an angle ϕ with the crank—that is to say, its phase is $\phi°$ before that of the crank (pars. 2 and 3).

Then the resultant horizontal force is $R_k \frac{v^2}{g\,r} \sin(\theta + \phi)$.

---

* In the following pages the inclusive minus signs are ignored.

Since the resultant is φ° in front of the crank, and also since the axis of reference is 90° in front of the vertical axis, the ordinate of maximum amplitude is $(\varphi + 90)^\circ$ in front of the ordinate B, and is, therefore, drawn through the point C. Divide the base line off again as before, but the number of divisions is the same as that of the complete circle, as there is here only one period per revolution. Draw the circle 4, 5, 6, and set off the ordinates as before. Then the curve D_4 G (fig. 220) is the required curve, the ordinates of which give the residual resultant horizontal unbalanced force at any position θ of the crank from the top centre.

Case 2.—Engine balanced perfectly horizontally. Here there is no residual horizontal force, and the curve therefore coincides with the base line, the amplitude always being zero.

In a vertical direction there is still the residual force $x \cos 2\,\theta$ as in case 1, but now there is an additional force due to the difference between the resultant of the weights required for vertical balancing and those that have been fitted for horizontal balancing.

Let W_1 W_2 be the weights as fitted, and W_h their resultant (fig. 222).

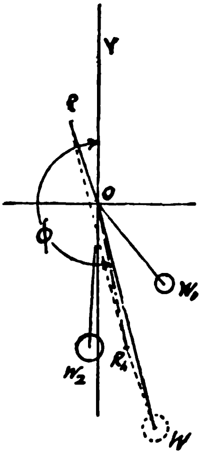


Fig. 221.

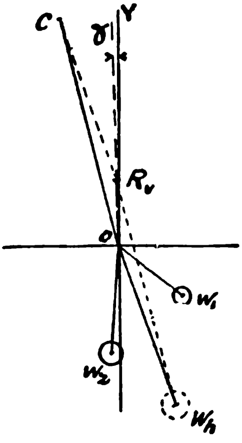


Fig. 222.

O C is the resultant of the reciprocating parts of the engine as drawn in fig. 211*a* (function of θ). Find the resultant of O C and $W_h = R_v$.

Then the resultant inertia force vertically is $R_v \frac{v^2}{g\,r} \cos(\theta + \gamma) + x \cos 2\,\theta$.

Although in Case 1 it was not absolutely necessary to draw the curve of force for comparative purposes, it is necessary here, because the maximum amplitude of the resultant is not the sum of the two amplitudes in the above expression.

Draw the curve $R_v \frac{v^2}{g\,r} \cos(\theta + \gamma)$, noting that its phase is γ° in front of the curve $x \cos 2\,\theta$.

Draw the curve $x \cos 2\,\theta$.

The resultant is the sum of the ordinates.

These curves are drawn in fig. 223.

The total amplitude H K is that required for comparative purposes.

This may seem on first reading a rather laborious process, but if the principles be thoroughly understood it will be found to be many times more rapid than drawing the curves by calculating the ordinates for various positions of the crank.

Case 3.—Engine balanced completely horizontally and vertically.

Here the residual horizontal force is zero, and the residual vertical force is the same as that in Case 1.

The curves just drawn are for a velocity, v, of the crank pin, corresponding to a certain number of revolutions, N, per minute.

The amplitude for any other number of revolutions, N_1, will be that just found multiplied by $\left(\frac{N_1}{N}\right)^2$.

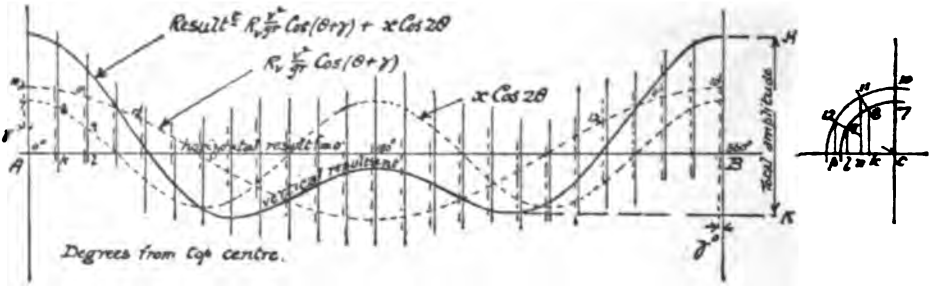


Fig. 223.

14. Curves of Free Force and Couples for a Multi-Crank Engine.—This will, as in the case of balancing (par. 11), be illustrated by means of the example of a three-cylinder engine—

Let M_1 = weight of H.P. reciprocating parts.
M_2 = „ M.P. „ „
M_3 = „ L.P. „ „

1. *Free Force.*—As the engine is as perfectly balanced as possible in both vertical and horizontal directions, the vertical free force will only be the resultant of the three residual forces $M_1 \frac{v^2}{gr\,s} \cos 2\theta$, $M_2 \frac{v^2}{gr\,s} \cos 2(\theta + 120°)$, $M_3 \frac{v^2}{gr\,s} \cos 2(\theta + 240°)$.

For convenience let the constants $M_1 \frac{v^2}{gr\,s}$, $M_2 \frac{v^2}{gr\,s}$, $M_3 \frac{v^2}{gr\,s} = x, y, z$ respectively.

Draw axes O X, O Y (fig. 224) and let the H.P. crank be along O Y.

In finding the resultant of the above three inertia forces or harmonic functions, *note particularly* that the angles between the generating lines are not the same as the angles between the cranks, because the forces are functions of 2θ. From A to B (fig. 225), which is 180°, really represents 360°, as far as the curves are concerned, and the sequence of the generating lines is as if the H.P. were leading instead of the L.P.

Then in fig. 224 the generating lines are in the order as shown. Find the resultant O R by the polygon as shown (par 3).

Then the resultant virtually leads the H.P. generating line by α°, or when referred back to the real degree scale (fig. 225) by $\frac{\alpha^\circ}{2}$.

The ordinate of maximum amplitude of the resultant therefore passes through the point C, and not through the point D, where it would if the generating lines were in the same sequence as the cranks.

Draw a quadrant of a circle of radius oR and set off the resultant curve as previously explained.

The engine being balanced perfectly horizontally the amplitude of the resultant force is zero, and the curve coincides with the base line.

If the engine were not completely balanced horizontally, the residual resultant horizontal force would, of course, be a function of θ, and the generating lines would have the same sequence as the cranks.

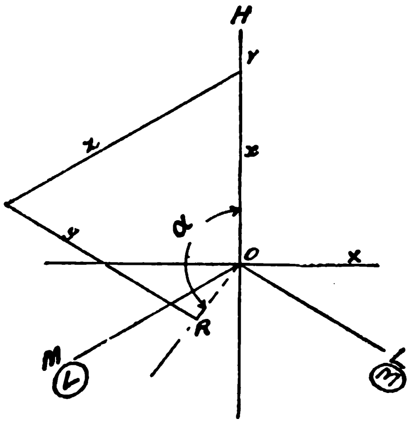


Fig. 224.

2. *Couples.*—As there are three centre lines, and a residual vertical force along each, there will be a residual resultant couple on the engine. The moment of the couple will, of course, vary according to the position of the reference plane, and in order to compare one design of an engine with another some common method of locating the reference plane is necessary.

The Admiralty, in their recent specifications, require it to be through a point on the shaft midway between the extreme cranks, and, considered generally, this is a very reasonable point to select, because it makes the length of the engine have a minimum influence on the value of the couple, and also causes the plane to be at about the same position in the ship, whatever the design of the engine may be. This point is thus suited for marine engines, but for land engines it is advisable to select a point that will enable a stricter comparison to be made. This point should be at what is termed

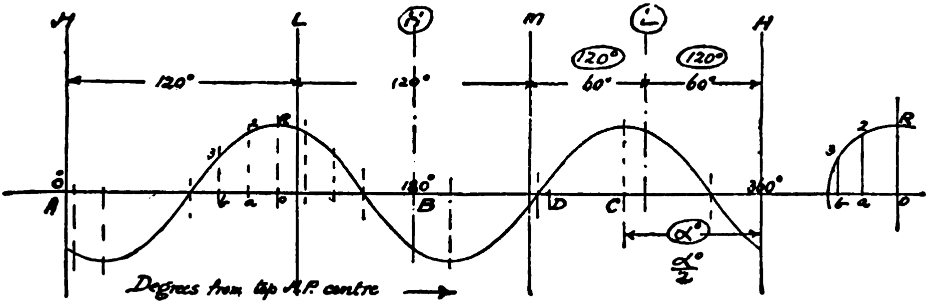


Fig. 225.

in theoretical mechanics "the central axis" of the couple. To explain this, let $F = OO$ be the resultant force (figs. 211*a* and 211*b*), and OC_1 the resultant couple referred to any arbitrary plane OY. Resolve the couple in and at right angles to the direction of the force F.

Replace the couple Oe—that is, in the same phase plane as the force—by another equal couple but having each of its forces equal to F. Thus $Fh = Oe$ (figs. 211*a* and 211*b*). Then F and - F on the reference plane cancel each other, leaving a force F on the new reference plane distant h from the original one and a couple eO_1 having its phase at right angles to the force and to the original couple.

Taking the case of the engine in par. 9, let the weights $P = 8$, $Q = 2$. Also, let $a = 4$, $b = 7$, and α as drawn.

$$\text{Then the resultant force } F = 7 \cdot 2\, x \cos(\theta + \delta)$$
$$\text{and the resultant couple } OC_1 = 27 \cdot 8\, x \cos(\theta + \omega).$$

Where $x = \dfrac{v^2}{g\,r}$ and neglecting the obliquity of the connecting-rod.

$$\text{The resolutes } Oe \text{ and } eC_1 \text{ are } 27 \cdot 2\, x \cos(\theta + \delta)$$
$$\text{and } 5 \cdot 8\, x \sin(\theta + \delta) \text{ respectively.}$$

$$\text{Then} \qquad \frac{27 \cdot 2\, x \cos(\theta + \delta)}{7 \cdot 2\, x \cos(\theta + \delta)} = 3 \cdot 78 = h.$$

Thus the central axis is 3·78 from the reference plane, and the resultants of the system are a force $7 \cdot 2\, x \cos(\theta + \delta)$ acting in the plane of the central axis or new reference plane, and a couple $5 \cdot 8\, x \sin(\theta + \delta)$.

The couple is always a minimum, and the central axis is, of course, always through the same point on the shaft, and therefore a reference plane which contains the central axis of the system is suitable for standard comparisons.

With this slight digression the consideration of the residual couples on the three-crank engine will now be proceeded with. The couple about an arbitrary plane must be taken first.

The moments of the residual vertical forces are—

$$M_1 \frac{v^2}{grs} a \,.\, \cos 2\theta \quad - \quad - \quad = x_1 \text{ say,}$$

$$M_2 \frac{v^2}{grs} b \,.\, \cos 2(\theta + 120^\circ) = y_1$$

$$M_3 \frac{v^2}{grs} c \,.\, \cos 2(\theta + 240\ ) = z_1.$$

Where a, b, c are the distances of the centres of engines from the reference plane respectively. Their signs must be according to the direction in which they are read.

Since the curves are all functions of 2θ the same remarks apply to the sequence of the generating lines in the polygon as for the free forces. The resultant may be found and the curve drawn in a similar manner.

If it be desired to find the couple about the central axis plane the process explained above may then be proceeded with. The phase of the resultant couple will then be 90° in advance of the original one, and will, of course, have a different amplitude.

As there is no horizontal force there is, consequently, no horizontal couple.

If the engine be not completely balanced for functions of θ in the vertical direction, then, in addition to the residual resultants above, the resultant of the "single period" (θ) forces and couples must be found separately and their ordinates added to those of the "double period" ($2\,\theta$) graphically, as explained in Case 2, par. 13.

15. The Effect of Inertia and Weight.—*General Case, Supplement to the Remarks in the Introduction to this Chapter.*—It has been shown that the vertical inertia force at any instant is $-\frac{M\,v^2}{g\,r}\left(\cos\theta + \frac{\cos 2\,\theta}{s}\right)$.

If this force is divided by the area of the piston the inertia pressure per square inch will be found. The effective pressure on the piston is therefore the algebraic sum of the steam and inertia pressures, and if the dead weight of the parts be included, the weight divided by the piston area must be added algebraically as well.

Considering the *inertia force* alone, M above is the total weight, including the rotating parts; but the revolving parts have also a horizontal acceleration, and the horizontal inertia force resolved tangentially to the crank path is equal and opposite to the vertical inertia force of the rotating parts resolved tangentially. This, of course, should be so, because the crank has no tangential acceleration. There is, therefore, no alteration produced in the effective twisting moment by the inertia of the rotating parts, and the weight of the reciprocating parts need only be used for obtaining the inertia pressure on the piston. M, in the above formula, will therefore be the weight of the reciprocating parts M_1.

It is easily shown that the twisting moment on the shaft due to an effective pressure on the piston P is—

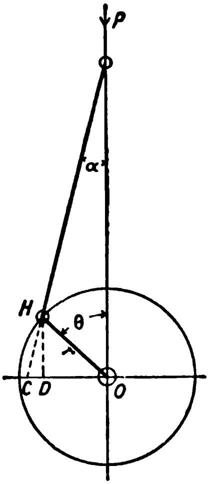


Fig. 226.

$$\mathrm{TM} = \frac{P\,r\sin(\theta + \alpha)}{\cos\alpha}, \quad \text{(fig. 226)},$$

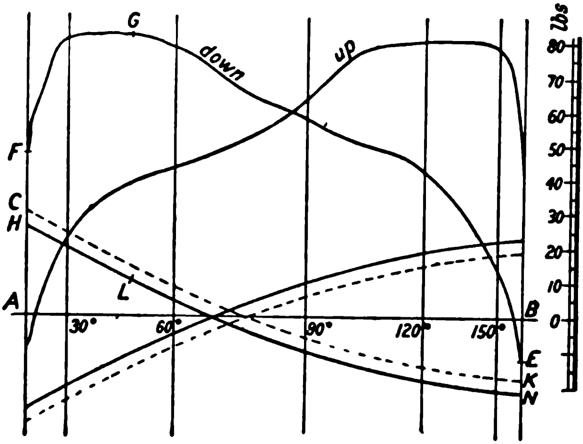
or, taking the second approximation to the series in terms of θ, as in the case of the acceleration (par. 4),

$$\mathrm{TM} = P\,r\left(\sin\theta + \frac{\sin 2\,\theta}{s}\right).$$

In order to combine the inertia curve with the indicator diagram or steam pressure curve, the same spacing of ordinates for each must be adopted. If the inertia curve be drawn with indicator diagram spacing ot ordinates (that is, the abscissæ are proportional to the distance of the piston from the end of the stroke) instead of abscissæ proportional to θ, it will assume the form shown in fig. 227*b* by the line O K. If $\cos 2\,\theta$ or the obliquity of the connecting-rod be neglected, the curve will be a straight line. The ordinates must of course be to the same scale as the steam pressure ordinates.

Considering the *dead weight* alone of the moving parts, the reciprocating

FIG. 227b



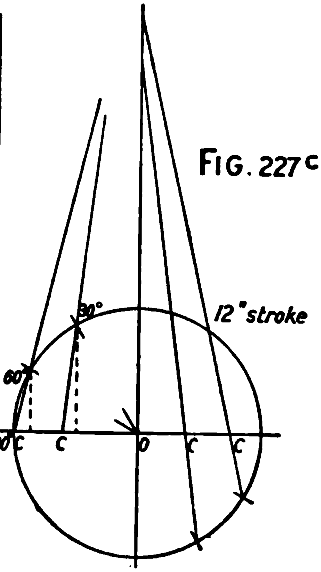
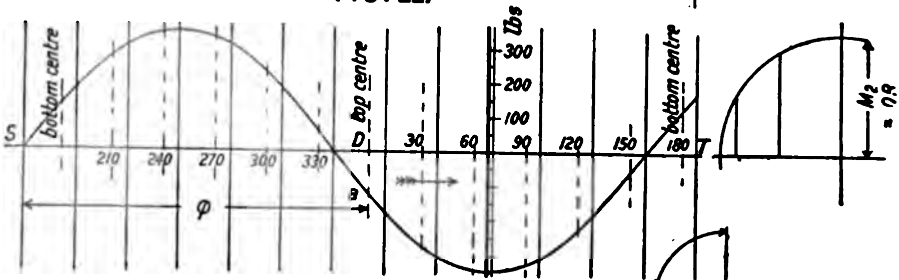


FIG. 227c

FIG. 227d



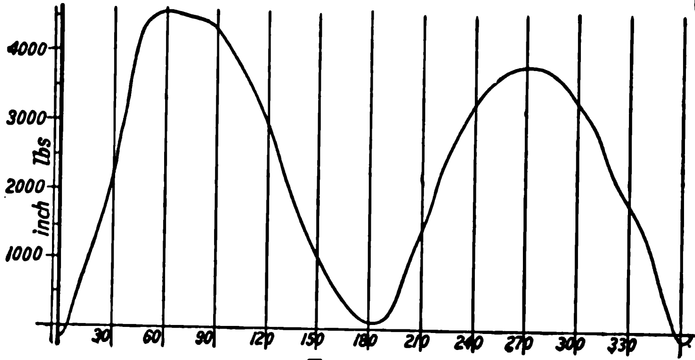


FIG. 227e

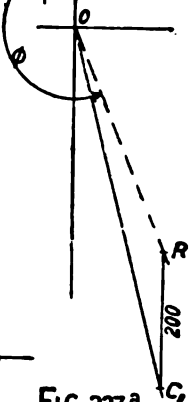


FIG. 227a

parts having their motion controlled by the connecting-rod must necessarily have their weight added algebraically to the inertia force. But the rotating parts also cause a twisting moment on the shaft which, if M_2 equal their weight at radius r, is equal to $M_2 r \sin(\theta + \phi)$, where ϕ is the angle between the resultant M_2 of all rotating masses and the crank. If there are no balance weights and valve gear be neglected, $\phi = 0$. M_2 not being controlled by the length of the connecting-rod cannot be considered as being distributed over the piston area (except for a connecting-rod of infinite length) and the effect of the weight of the rotating parts must therefore be considered apart from the indicator diagram.

Then (fig. 227*b*) the weight of the reciprocating parts being a constant quantity, set off $HO = \frac{M_1}{A}$ downwards for the downstroke and upwards for the upstroke, and draw a curve, H N, through H, parallel to the inertia line, O K.

Set off the indicator diagram from the straight back pressure line, A B.

Then the effective pressure on the piston at the beginning of the stroke is not A F but H F, and at any other position of the piston the effective pressure, P, is given by the intercept between the curves F G E and H N. And similarly for the upstroke, due note being taken of the algebraic sense of the various pressures.

The total effective twisting moment at any instant is now—

$$TM = Pr\left(\frac{\sin(\theta \times \alpha)}{\cos \alpha}\right) + M_2 r \sin(\theta + \phi).$$

$Pr\frac{\sin(\theta + \alpha)}{\cos \alpha}$ may be calculated in a semigraphical manner as follows:—

Let the crank be at O H (fig. 226), then

$$Pr\frac{\sin(\theta + \alpha)}{\cos \alpha} = P \times OC.$$

Also, when

$$\phi \text{ is } 0, \text{ then } M_2 r \sin\theta = M_2 \times DO.$$

16. An example of Case 1, p. 553, will now be considered.

Although it is very necessary to include the valve gear when balancing, it is rather an unnecessary elaboration to consider its influence on the twisting moment diagram as it is so very small, unless, of course, the gear be abnormally heavy :—

Let M_1 = 400 lbs. (reciprocating).
M_2 = 200 lbs. (rotating).
Revolutions = 200 per minute.
12-inch cylinder, 12-inch stroke.
Connecting-rod 2 feet centres.
Position of fittings on shaft as in fig. 211.

Then

$$v = 10{\cdot}47 \text{ feet per second.}$$

$$v^2 = 109{\cdot}6$$

$$s = \frac{24 \text{ inches}}{6 \text{ inches}} = 4$$

$$\frac{M_1 v^2}{A g r} = 24{\cdot}1.$$

The values of the inertia pressure, &c., for every 30 degrees of the crank are tabulated below (fig. 227, *a*, *b*, *c*, *d*, *e*):—

	$-\frac{M_1 v^2}{A g r} \times \cos\theta + \frac{\cos 2\theta}{n}$.	P = G L.	O C (Fig. 227*b*).	P . O C × A = Q_1.	$M_2 \times \sin(\theta + \phi)$.	$M_2 \sin(\theta + \phi) \times r = Q_2$.	T M = $Q_1 + Q_2$.
	Lbs.	Lbs.	Inches.	Inch-lbs.	Lbs.	Inch-lbs.	Inch-lbs.
0°	− 30·1	...	...	...	− 130	− 780	− 780
30°	− 23·9	60	3·64	24,700	− 270	− 1,620	+ 23,180
60°	− 9·04	73	5·82	48,000	− 345	− 2,070	+ 45,930
90°	+ 6·02	67	6	45,500	− 330	− 1,980	+ 43,520
120°	+ 15·06	60	4·52	30,660	− 225	− 1,350	+ 29,310
150°	+ 17·85	36	2·35	9,580	− 60	− 360	+ 9,220
180°	+ 18·08	...	...	...	+ 130	+ 780	+ 780
210°	+ 17·85	56	2·35	14,860	+ 270	+ 1,620	+ 16,480
240°	+ 15·06	61	4·52	31,200	+ 345	+ 2,070	+ 33,270
270°	+ 6·02	54	6	36,700	+ 330	+ 1,980	+ 38,680
300°	− 9·04	48	5·82	31,600	+ 225	+ 1,350	+ 32,950
330°	− 23·9	43	3·64	17,700	+ 60	+ 360	+ 18,060
360°	− 30·1	...	...	...	− 130	− 780	− 780

To find ϕ, let O C_1 (figs. 211*a* and 227*b*) be the resultant of the balance weights and O_1R equal weight of rotating parts (200 lbs.) then O R is their resultant.

Draw the curve O R sin θ (fig. 217*b*).

Set off S D = ϕ, then D*a* = O R sin ($\theta + \phi$), or M_2 sin ($\theta + \phi$).

Fig. 227*e* is the effective twisting moment diagram.

17. The area of the final twisting moment diagram is proportional to the work done, and although its form has been modified by the inclusion of the inertia and weight, the area remains the same as if the steam pressure alone were considered.

This also follows from the principle of conservation of energy.

If the engine be balanced completely horizontally, and not vertically as in Case 1, the inertia and weight pressures from the reciprocating parts are the same as above, but the moment of the weight of the rotating parts is zero at all positions of the crank.

If the engine be balanced by bob weights (two) then the inertia of these must be considered exactly as if there were two extra cylinders in the engine, the only difference being that there is no steam diagram to combine with their inertia diagram.

The foregoing process applies equally well to a multi-crank engine.

The reciprocating weights and inertia pressures must be found separately for each cylinder, and the resultant horizontal weight-twisting moment for the whole engine combined with the resultant twisting moment due to the effective piston pressures.

18. **Stresses due to Inertia.**—As the stress in a given piece of machinery is due to the effective load applied to it, it therefore follows that inertia and dead-weight loads should be added algebraically to the steam loads when finding the stresses in those parts that are subject to them.

In very slow-running engines the effect of inertia is inappreciable, and may therefore be neglected, but in quick revolution and high-piston-speed engines its effect must not be lost sight of.

Starting at the piston, the first section to be dealt with is through the piston-rod thread. The inertia force here is that due to the weight and acceleration of the piston and not of the whole of the moving parts. The stress in the thread only comes into play on the upstroke, and the effective pressure on the piston is the intercept between the steam and weight lines as shown above.

Similarly, the stress at any section between the top of the piston and the centre of gravity of the connecting-rod may be found, the inertia line being constructed from the weight of the parts above the section.

As a rule, the factor of safety allowed for marine engines of the ordinary type amply covers the additional stresses due to inertia, and an engine that is well designed for static stresses may generally be considered to be strong enough to provide for dynamical stresses as well.

Referring to fig. 227*b*, it will be observed that the inertia line tends to make the effective pressure constant throughout the stroke.

For engines of normal scantlings and speed the maximum effective pressure usually occurs at about two-thirds the stroke from the beginning.

As the weights of moving parts are rarely available when an engine is being designed, and as there are also no indicator diagrams to refer to, the following approximate formula for finding the maximum effective pressure is given. This formula includes the total weight of reciprocating parts.

Let p_m = mean pressure of steam for that cylinder from the proposed I.H.P.,
p_f = maximum effective pressure,
L = length of stroke in feet,
N = revolutions per minute,

then

$$p_f = p_m + \cdot 0000386\ N^2 L^2.$$

The formula applies to the low-pressure cylinder for engines where there is only one low-pressure cylinder. If there are two low-pressure cylinders, then the medium pressure should be taken. It is not suitable for high-pressure cylinders of compound expansion engines, but this does not detract from the value of the formula, because the maximum loads generally occur in the above mentioned cylinders.

Example.—To find the maximum effective pressure that may be expected on the low-pressure piston of a triple-expansion engine of 1500 I.H.P., 150 revolutions, 54-inch low-pressure cylinder, 2 feet stroke—

$$\begin{aligned} p_m &= \frac{1500 \times 33{,}000}{3 \times 2290 \times 2 \times 2 \times 150} \\ &= 12 \cdot 00 \text{ lbs. per square inch.} \end{aligned}$$

$$\begin{aligned} p_f &= 12 \cdot + \cdot 0000386 \times 22{,}500 \times 4 \\ &= 12 + 3 \cdot 47 \\ &= 15 \cdot 47 \text{ lbs. per square inch.} \end{aligned}$$

Example.—To find the diameter of the connecting-rod bolts (two) for the above engine, allowing a stress of 5000 lbs. per square inch—

Maximum working load on piston = 2290 × 15·47 = 35,426 lbs.

$$\begin{aligned} \frac{35{,}426}{2 \times 5000} &= 3 \cdot 54 \text{ square inches at bottom of thread.} \\ &= 2\tfrac{1}{2}\text{-inch bolts.} \end{aligned}$$

19. The Design of Balance Weights.—As a rule, and more particularly with two and four-crank engines, the great difficulty to contend with is being able to get the required mass in the space available.

With a uniform thickness and external radius there is a limiting length of weight (circular arc) beyond which any increase of mass makes the total mass less effective. The limit varies according to the shape of the weight and the angle which it makes with the crank.

When the weight is homogeneous and fairly opposite the crank its length should not subtend a greater angle than 120°, and in no case more than 140°, at the centre of the shaft. When inclined at a considerable angle to the crank, as shown in figs. 228 and 229, its length should not subtend more than 90°, on account of the long arm for attaching to the crank web—for any extension of length beyond these limits causes the centre of gravity to approach the centre of the shaft at a greater rate than the mass is increased.

For composite weights, as in figs. 228 and 229, the angles subtended with the centre of shaft may be increased about 5° above those given above.

The material of which balance weights are made depends, to a great extent, on the manner in which they are attached to the crank-webs.

In small engines, and when the weights of the moving parts are obtainable at the time the crank shaft is designed, and when the crank-shaft is a built one, the weights are very often forged as part of the webs.

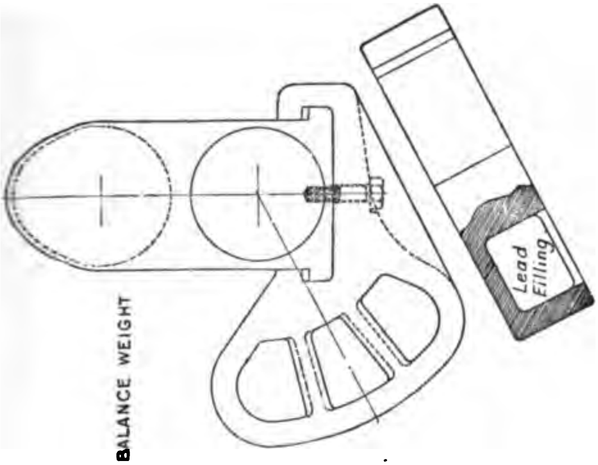
Circumstances, however, generally necessitate loose balance weights being fitted.

A neat and compact method of attachment is shown in figs. 228 and 229, and consists of forging small lips on the crank-webs, over which the weights are keyed.

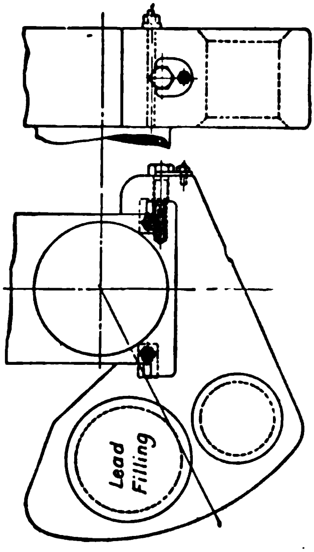
The material the weights are made of, when this method of attachment is adopted, must necessarily not be of a short nature, and should either be forged or cast steel of a low tensile strength.

The Admiralty and a few other owners will only accept forged material, but there is no reason why a suitable cast steel should not be used for them. In fact, it has been used successfully by the author. The advantage of a cast weight lies in being able to make it in the form of a shell which may be filled with lead, the materials thus being most advantageously disposed and the weight being of the smallest possible dimensions. It is also, generally, less expensive. This is shown in fig. 228. A forged balance-weight is shown in fig. 229. The most convenient method of manufacture is to shape them with a band-saw out of rolled slabs. Parts of the forging may be drilled out and filled with lead as shown, but at best this is an expensive addition and hardly worth the extra labour, as the quantity of lead that can be introduced is necessarily only a small fraction of that in fig. 228. Cast-iron weights are used to a great extent on high-speed land engines, and to a less extent for marine work. A cast-iron weight cannot be relied on if held on by lips as in figs. 228 and 229, and the usual method of attachment is by strapping it on as shown in fig. 230. The casting may be cored out and filled with lead if the small space available for it makes it advisable.

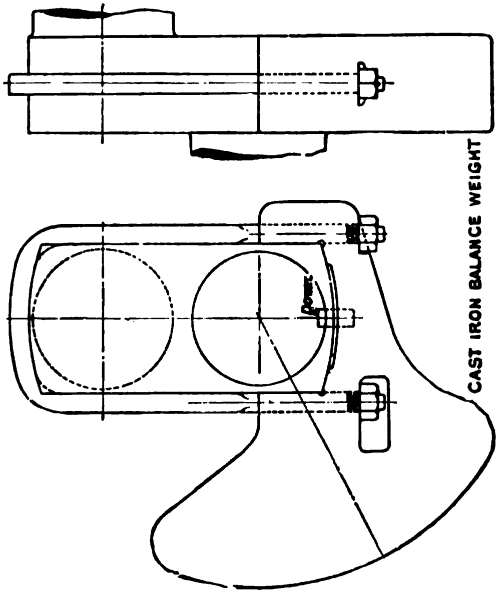
Cast-iron weights with its straps and fittings are, on the whole, much cheaper than either cast or forged steel, and there does not seem to be much reason why, beyond being somewhat clumsy in appearance, they should not be used to a greater extent than they are.



CAST STEEL BALANCE WEIGHT



FORGED STEEL BALANCE WEIGHT



CAST IRON BALANCE WEIGHT

Figs. 228, 229, and 230.

In all cases the lips or straps must be made strong enough to take the centrifugal force of the balanceweight, together with the deadweight itself. If strong enough for this they will, in general, be found strong enough to resist the impulsive force on them, due to the engine being suddenly started or stopped.

It is not expedient in a manual of this kind to devote more space than the foregoing to the problem of balancing. Literature on this subject is exceedingly voluminous and fairly accessible. The aim of this chapter has been to place the elementary and practical principles before the student in such a manner that, should he desire it, he would have no difficulty in pursuing the subject more profoundly from the many highly mathematical papers extant.

For instance, slight mention only has been made of the Yarrow-Schlick-Tweedy system of balancing. This is a system only applicable to an engine with not less than four cranks, and the essential feature is that the weights of moving parts in conjunction with the position of cranks are so arranged, that the parts balance, or nearly balance, themselves without the introduction of heavy weights.

Sometimes small rotating balance weights are fitted to the cranks, their special purpose being to assist in balancing residual horizontal forces and couples, it being difficult to arrange a complete vertical and horizontal balance by the various moving parts *inter se.* The student will easily recognise that any two of the piston lines are really acting as bob weights for the other lines, and he will have no difficulty in applying the principle of closed polygons already described to the forces and couples.

By this system it is also possible to balance the secondary forces and couples (functions of 2θ) as well as the primary, and there have been several exceedingly ingenious geometrical methods of doing this published. Their great drawback, however, is that, as a rule, they neglect valve gear, and it is labour wasted to treat part of the problem by a refined mathematical process, because of its susceptibility to exact calculation and ignore parts comparatively massive, such as the valve gear of the modern marine engine, which seem to be far more important.

Solutions are to be found including all the elements (with necessary assumptions such as given on p. 556); they are necessarily complicated, but with a fair amount of patience and scheming they may be arrived at.

The methods, short cuts, &c., of attacking the problem will suggest themselves, as he proceeds, to the designer who has got a thorough grasp of the principles involved.

CHAPTER XXV.

MATERIALS USED BY THE MARINE ENGINEER.

In every ship the weight of the machinery is of some consideration, and in many ships it is of the utmost importance that it be as light as possible. Reduction in weight can only be made by using material of superior strength, and by so designing the engine as to be able to employ such materials. The most marked advances in marine engineering, both in the size and quality of the work turned out, are in great measure due to the superior materials now available, and to the means possessed for converting them to the engineer's requirements. Steel, which was, only a few years ago, a comparatively expensive material, has now practically displaced iron, and that in most instances on the score of cheapness. Not only is the price of steel now so low that it can compete successfully with iron, but steel can be made in such large masses at such comparatively small cost, that heavy shafts made of this material render possible the immense engines recently constructed for the large ocean steamers. Now steel alloys, especially nickel steel, are being made of such excellent quality, and with such high elastic limits, that for high-class engineering they are likely to displace ordinary mild steel, notwithstanding their prices being so much higher. Bronzes, too, have commanded attention, for not only do they hold their own for special purposes, but since certain kinds can be manufactured with as high a tensile strength as steel, and stand hammering hot or cold, they have displaced steel and iron for such purposes as propeller blades, parts of the engines, &c., in spite of the very high price.

Cast Iron.—Although steel has superseded iron in many ways, cast iron still remains as the material most largely used by engineers. Its appearance is so well known as to need no description. The qualities of cast iron are very numerous, and are generally known by the district from which the ore is obtained. There also still remains a distinction from mode of manufacture, as "cold blast" is distinguished from "hot blast" in other ways than that of cost.

All the pig iron of each district is, as a rule, divided into seven qualities, each of which is known by a number; that containing most free carbon, or *graphite*, is designated No. 1, that containing least No. 7.

No. 1 pig, when broken, exhibits a very coarsely granular fracture, having dark grey scales of considerable size; when melted "it runs very thin," that is, becomes extremely fluid, and on that account is used for ornamental castings and other work which requires a sharp outline, or is of very little thickness. It is not used much by marine engineers, except to mix with other kinds, or where extreme fluidity is necessary. Castings made from it are very soft unless they are so thin that the metal has changed its character in the mould by chilling. The carbon here exists chiefly in the form of graphite scales, mechanically mixed as it were with the iron crystals.

No. 2 pig is not so soft nor so fluid when melted as No. 1, but is still not sufficiently close grained for general use.

No. 3 pig is that usually employed by the marine engineer for general purposes, as by adding some of No. 1, a mixture suitable for a complicated casting is obtained, and by adding to it some of No. 4, a harder and closer grained casting can be made.

No. 4 pig is not much used for foundry purposes, except as a means of closing the grain and hardening the metal. It also differs from the other numbers in the appearance of the fracture; they all present a highly crystalline fracture of a distinct grey colour, and on that account called *grey iron;* it shows as a grey iron at the fracture, but the grain is much finer, and there is an absence of the coarse graphitic scales so strongly marked in the other three numbers.

Nos. 5 and 6 are not used at all for foundry purposes, but made for manufacture into wrought iron; on this account they are called *forge irons*, and as the fracture is still somewhat grey in places it is called "grey forge," to distinguish it from *No. 7*, which is called "white forge," as the fracture displays a distinctly crystalline structure, very hard, and silvery white in appearance. The carbon in this case is wholly combined with the iron chemically. Nos. 5 and 6 often present a mottled appearance, as if a grey iron and a white iron had been melted and imperfectly mixed; it is on this account that it is often known as "mottled pig."

It is often customary to generalise and recognise only three varieties of pig iron, viz.:—No. 1 as *grey* iron, No. 2 as *mottled* iron, and No. 3 as *white* iron; Nos. 2 and 3 being, of course, looked on as "forge" iron, and No. 1 as "foundry" iron.

In Great Britain there are many kinds of pig iron used by moulders, each called by the district where the ore is raised and smelted.

Scotch Iron is deemed the best for foundry purposes; it is very uniform in quality, of good strength, and will mix well. There are various brands, those of Gartsherrie, Glengarnock, Eglinton, Carron, &c., &c., being best known. The No. 3 pig is most generally used by marine engineers, as it will run sufficiently fluid to make any casting, and, judiciously mixed with scrap, it can be depended on for both closeness of grain and strength.

Cleveland Iron is used very much in the Cleveland district for general work; it is harder than Scotch iron, but does not possess so much strength, nor is it nearly so tough. A mixture of Nos. 1 and 3 is used for large work requiring strength, and this, when melted, possesses fluidity enough for the ordinary marine castings. Cleveland iron alone is not fit for large marine cylinders, columns, &c., but with the addition of hæmatite and scrap, to a mixture of Nos. 1 and 3, a good casting is obtained for general purposes.

Lincolnshire Iron is about equal in quality and general description to the Cleveland.

Staffordshire, Yorkshire, Derbyshire, and **Welsh Irons** are very good, and used in their own districts generally.

Shropshire Irons are very good, and from this county, Monmouth and Stafford the famous cold blast irons are obtained.

Cumberland Iron is made from *hæmatite* ore, and the pig generally goes by the name of "hæmatite." It is generally used for steel-making, but is also employed to improve other irons for foundry use. It possesses great strength and toughness, but cannot be used by itself for foundry purposes, as it does not run well when melted; 20 per cent. blended with good Scotch and English irons makes a strong mixture.

Cold Blast Iron.—Iron manufactured with air at the ordinary temperature is called by this name to distinguish it from the generality of irons which are made by heating the blast to about 700° Fah. The best known and most generally used cold blast irons are "Blænavon" and "Lilleshall," which owe part of their excellency, no doubt, to the good ore from which they are made. This iron possesses great strength with closeness of grain, and is used to close the grain and strengthen other irons.

Iron Mixtures.—All cast iron is improved by re-melting, and the improvement continues until it has been re-melted as many as twelve times; after this it falls off in strength. No important casting should, therefore, be made from new iron only, and such as cylinders, pistons, cylinder covers, &c., should be made from iron wholly re-melted if the utmost strength is to be obtained. This rule, however, is only carried out when the thickness of metal is cut down to the lowest limit to save weight.

Cylinders, &c., which are subject to shock as well as to changes of temperature and severe strains, should be made of strong tough metal, and, since a good surface is required to withstand the wear of pistons and valves, the metal must have a close grain. Such cylinders, &c., should be made of a mixture of one-third picked scrap, one-third best Scotch No. 3 pig, and one-third Blænavon or Lilleshall. If the casting is not large and complicated, the grain may be closed by adding picked scrap of a hard nature and increasing the amount of Blænavon. Some moulders add *hæmatite* to this mixture to increase the strength, but this is not often done, as it decreases the fluidity.

If the cylinder is to have liners and false faces for the valves to work on, it needs only to be *strong;* all *hard* materials therefore may be omitted.

Cylinder liners and false faces require to have a fair amount of strength, and be as hard as consistent with capability of being machined. Scrap iron of close grain and hard nature is selected to add to best Scotch No. 3 pig and Blænavon; if hard scrap cannot be obtained, some No. 4 pig may take its place; if necessary, even a small portion of "white" iron may be added to give additional hardness. Hardness and strength with closeness of grain may be obtained by mixing scrap steel (shearings and punchings from boiler-plates) to the extent of even 10 per cent. As the metal becomes much stiffer on the addition of the steel, only thick plain castings can be made with it. Propeller blades and bosses may with advantage be made with a mixture containing steel in lieu of hæmatite; the quantity of steel which may be added for this purpose depends on the founders being able to melt it.

Foundations and other large masses which are necessarily heavier than absolutely needed for strength may be made of a poorer mixture than suffices for cylinders; but as they are liable to shock, the iron must be of such a description as to resist this.

Specific Gravity of Cast Iron varies from 6·886 to 7·289; the average may be taken at 7·20. The weight of a cubic foot is, therefore 450 lbs. and a cubic inch 0·26 lb. A square foot of it 1 inch thick weighs 37·50 lbs.

Strength of Cast Iron.—The following is the result of some careful experiments made at Woolwich Arsenal some years ago:—

	Minimum.	Maximum.	Average.
Tensile strength per square inch,	4·85 tons.	14·05 tons.	7·36 tons.
Transverse „ „ „	1·37 „	4·47 „	...
Torsional „ „ „	1·74 „	3·44 „	...
Crushing „ „ „	22·54 „	58·42 „	50·00 „
Shearing „ „ „	...	...	12·00 „

Good cast iron made by mixing qualities suitable for marine work should have an ultimate tensile strength of 25,000 lbs. and resistance to crushing of 110,000 lbs.; a bar 1 inch square and 36 inches long should deflect ½ inch without breaking with a load of 800 lbs. at the middle.

Wrought Iron.—This material, which is very nearly pure iron, is made exclusively in this country by the "indirect process"—that is, manufactured from cast iron by the process called puddling. The old methods of obtaining malleable iron direct from the ore, hence called the "direct process," are practised only by semi-savage tribes, or in regions inaccessible to general trade. The Siemens method of manufacturing steel is to some extent a reversion to the "direct process," and the product is sometimes properly called *ingot* iron, to distinguish it from iron made from the "*bloom*" which is obtained by puddling.

Wrought iron is used by engineers in many forms, the principal of which are bars, plates, and forgings.

Rolled Bar Iron.—The "blooms" from the puddling furnace, after being squeezed or hammered, are rolled into bars, which are known as "*puddled bars;*" this bar iron is not used by engineers, as its tensile strength is sometimes as low as 9 tons per square inch; it is cut into pieces, which are piled crossways into a "faggot" or "pile," reheated, and rolled again into bars, and now called "*merchant bar*" iron.

Merchant Bars are not used by engineers for any very important work, as the iron is still of low tenacity and not very uniform in structure or quality. It was used for making gratings, ladders, &c., for fire-bars, bearers, &c.

Best Bar is made by reheating "faggots" of merchant bar iron, or good wrought-iron scrap "cross piled," and rolling it again into bars. Its strength is now much improved, and its quality more uniform, and it may be used for general smithing purposes. Its tensile strength, if made of good material, is about 24 tons per square inch on the average; the "Best" bars of some makers will withstand as much as 26 tons per square inch.

Best Best Bar is made by again rolling bars from faggots of selected Best bars. It has now a very uniform silky fibre, will bend double cold, and has a tensile strength of 26 to 27 tons; good specimens should elongate 25 per cent. at fracture with a reduction of area of about 50 per cent. Some kinds of iron have even a higher tensile strength than this, especially when rolled into round bars. Bar iron by cold rolling is increased in strength, but the elongation is reduced. Such iron is, however, seldom or never used by engineers.

Rivet Iron should be soft and of very good quality to withstand the work put on it in the process of riveting; its tensile strength is therefore somewhat low, about 24 tons per square inch, and its resistance to shearing is even lower than this, being from 20 to 22 tons per square inch. This iron is generally made by rolling "piles" of selected scrap iron into bars, and for best quality rivet iron, the bar is rolled from "piles" of ordinary rivet iron.

Rolled Scrap Iron.—A superior quality of bar iron is made by "piling" shearings from boiler plates and rolling in the usual way; the quality of this iron, however, depends very much on that of the plates from which the shearings came.

Weight of Bars.—The specific gravity of bar iron of good quality is on the average 7·62; the weight of a cubic foot is therefore 476 lbs., and that of a cubic inch 0·276 of a pound. Yorkshire bar iron is somewhat denser, its specific gravity being 7·76; so that a cubic foot of it weighs 485 lbs.

Angle Bar Iron.—This was formerly an important form of bar for boiler-making, but is now seldom used for that purpose; it is, however, still used sometimes in the construction of smoke-boxes, uptakes, &c.; but ship quality steel has practically taken its place.

Yorkshire Iron.—The best kinds of boiler iron were made in South Yorkshire, in the neighbourhood of Bradford and Leeds, and known as "best Yorkshire iron." Krupp, and some other German manufacturers, made iron plates of a quality equal to this; Swedish and Russian plates are very similar to it, and in some respects of superior quality.

It was of very uniform quality, and, although not possessing a very high tensile strength in direction of the grain, it was superior to other irons in strength across the grain; it was very tough, and had great elasticity, so that it was easily flanged and bent, and stretched very considerably before breaking. For these reasons, it was most valuable for boiler-making, and notwithstanding its high price (on the average three times that of ordinary boiler plates) it was always used for those parts of a marine boiler exposed to flame. Bar iron of this quality is still made in considerable quantities for smithing purposes in railway work, as well as for the screwed stays of boilers. It is practically as strong as boiler steel, and stands alternating stresses better than ordinary steel; it does not, however, take so good a screw thread. It is also used for crane chains and other high-class work, and much esteemed for colliery work.

Best Yorkshire plates had a tensile strength of 24 to 25 tons with the grain, and 22 to 23 tons across it; the elongation being 13·5 and 8 per cent. respectively. From some carefully made experiments, Mr. Kirkcaldy found the average strength of Yorkshire iron to be 21·3 tons with and 20·1 across the grain, the elongation being 16·7 and 11·2 per cent.; by annealing the plates the strength was slightly *reduced*, but the elongation was raised to 18·4 and 12·8 per cent. The elastic strength was also found to be 12·2 tons in tension, and in compression 11·5 to 13·3 tons.

Its specific gravity is 7·76; the weight of a cubic foot is 485 lbs., and that of a cubic inch is 0·281 lb.

Staffordshire Iron.—This iron, although slightly inferior to best Yorkshire, is still of high quality, and was used for boiler shells, domes, &c., and for such parts of the furnaces and chambers as were not exposed to the direct action of flame. It has a high tensile strength with the grain, but is not so strong across the grain as is the Yorkshire iron. Sir William Fairbairn, in 1861, found that some Best Best Staffordshire plates had an ultimate strength of 26·7 tons with and 24·47 tons across the grain, the elongation being 6·7 and 4 per cent.; that common Staffordshire plates had an ultimate strength of 22·7 tons with and 23·5 across the grain, the elongation being 5 and 4·35 per cent. The large plates of Best Staffordshire quality, as now rolled, are found to have an ultimate strength of 23 tons with and 19 across the grain—it is only the thinner plates that possess so high a strength across the grain; the ordinary qualities of Staffordshire boiler plate, when ¾ inch and upwards in thickness, rarely have a tensile strength over 18 tons across the grain.

For purposes of calculation, the ultimate strength of Staffordshire quality boiler plates may be taken at 51,500 lbs. with, and 43,000 lbs. across, the grain for plates under ¾ inch thick; and at 50,000 lbs. with, and 40,100 lbs. across, the grain for plates over that thickness.

The specific gravity is 7·68, the weight of a cubic foot being 480 lbs., and that of a cubic inch 0·277 lb.

Staffordshire Bar Iron is very largely used for chains and similar gear, as well as for smithing purposes, and is very highly esteemed for them.

Cleveland Iron.—As plates are made in North Yorkshire, and in various parts of Durham, from this iron, it is better to deal with it under this name, rather than by the name of the district in which the plates are rolled. The plates formerly sold for shipbuilding and bridges are far inferior to those of Staffordshire make; they were hard and brittle, and required careful usage; from its containing considerable proportions of phosphorus, this iron is "cold short," that is, liable to fracture across when worked cold; for this reason all serious operations on it must be conducted while it is hot, as even punching very often causes it to crack from the hole, and plates which have been punched will break through the holes while being bent in the rolls.

The ultimate tensile strength of this iron is only at best 22 tons with and 18 across the grain, and the commoner descriptions of boiler plate have an ultimate strength of 20 to 21 tons with and 17 tons across the grain; and the elongation is very small.

This was formerly used by engineers for smoke-boxes, funnels, and casings, &c.; now ship quality steel is employed as being cheaper and easier handled.

Iron Forgings.—Iron forgings, such as shafts, rods, &c., are made from scrap iron, and their strength depends very much on that of the iron from which the scrap was cut. Sometimes forgings are made from new iron, but, since the general use of steel, this is seldom done now. The method of manufacture is similar to that described for making rolled bars from scrap; the scrap is sorted, piled, brought to a welding heat, and hammered into slabs; the slabs are piled one on the other, and reheated to form the forging required. The best description of forging is made by rolling the slabs into bars, so as to give the metal *grain;* the bars are then cut into short lengths, piled, and hammered again into slabs, which are piled, &c., as before, to form the forging. This rolling into bars, in addition to giving the iron fibre, tends to give a more uniform structure to the forging, and a homogeneity which cannot be obtained by the simple piling process.

The strength of forged iron depends also on the extent to which it has been worked under the hammer, and it is no doubt also considerably affected by the heat to which it has been exposed during the forging. The weakness of some forgings is often due to the continuous exposure to very high temperature in the reheatings, so often necessary in producing a heavy complicated piece of work.

From some experiments made by the Mersey Steel and Iron Company on portions of iron cut from a crank-shaft, the tensile strength was found to be 22·4 tons, with an elongation of 25 per cent. The iron in this case was, no doubt, exceptionally good and the forging sound. The strength of forgings made from good scrap iron is probably 22 tons per square inch with the grain, and 19 tons across, when they are of simple form and moderate size; when of large size, or of complicated form, like a crank-shaft, the strength seldom exceeds 21 tons with and 18 tons across the grain. The elongation with good iron should be at least 15 per cent. with and 8 per cent. across the grain.

The specific gravity of large forgings is about 7·63, so that the weight of a cubic foot is 477 lbs., and that of a cubic inch is 0·276 lb.

Steel.—Steel, like iron, is used in the form of bars, plates, and forgings, and is also very generally employed for castings where great strength is required.

All steel was originally made from the best qualities of wrought iron by the process of "cementation;" this consists of exposing small pieces of iron to a high temperature in the presence of carbon only in a closed vessel for a considerable time, during which some of the carbon is absorbed by the iron, and is thus converted into a rough kind of steel, called *blister steel.* These pieces are broken, and sorted according to the appearance of the fracture, after which portions are placed in a closed crucible, melted, and cast into ingots; it is now called *cast steel.* If the blister steel is piled, reheated, and hammered, or rolled into bars, it is called *shear steel.* The cast-steel ingots are rolled into bars, which still retain the name "cast steel," but this is better known as *tool* steel, as it is now used almost exclusively for cutting-tools. Tool steel containing about 1 per cent of carbon is, of course, very hard, and has a very high tensile strength, ranging from 50 to 65 tons per square inch, with an average elongation of a little over 5 per cent. only; some of the milder kinds, such as are used for drifts, &c., have a tensile strength varying from 44 to 60 tons per square inch, with an average elongation of 13 per cent. *Spring steel* is still milder, having a tensile strength of about 45 tons per square inch, with an elongation of 18 per cent. Tempering in oil increases the strength considerably; Mr. Kirkcaldy found that a steel bar, whose strength when "soft" was 54½ tons, when heated and cooled in oil had a strength of 96 tons.

Bessemer Steel.—The modern methods of making steel known as the *direct processes* obviate the necessity of using the comparatively expensive wrought iron, and are therefore capable of producing a very much cheaper material. In the Bessemer process there are essentially two operations, the conversion of molten *cast* iron into *pure* iron, and, by the addition of a small and definite quantity of manganese and carbon, the turning of pure iron into steel. Cast iron practically free from phosphorus and sulphur is melted and poured into the converter; a strong blast of air is forced through the molten metal, so that the carbon it contains burns and is consumed, and the temperature of the mass thereby raised; when the whole of the carbon is consumed, a small quantity of *ferro-manganese* (spiegeleisen or natural ferro-manganese was used originally), an iron containing a known proportion of carbon and manganese, is added; the metal now has that small amount of carbon which causes it to differ from wrought iron, and that amount of manganese which seems to be so essential in making good steel. The metal is now run into ingot moulds, and allowed to cool for further use, or is kept in a "soaking pit," reheated and hammered, or rolled into the forms required. By the Thomas-Gilchrist process fairly good steel can be made from iron containing phosphorus; the phosphorus is absorbed by the converter lining, which is prepared from magnesian limestone; the product is known as *basic* steel, that made by the original Bessemer process being called *acid* steel.

Siemens-Martin Steel.—The steel in this process is made in the hearth of a reverberatory furnace by exposing for a considerable time hæmatite or pig iron, or mixtures of cast iron and high-class pure (oxide) ores of iron, to the intense violet heat obtained by producer gas until practically the whole of the carbon disappears. If pig containing phosphorus is used, the furnace must be lined with *gannister* made from magnesian limestone on the Thomas-Gilchrist plan, so that the phosphorus may be absorbed almost wholly by the lining during the roasting process; *ferro-manganese* is added as in the Bessemer process, and the liquid mixture run into ingot moulds.

Steel made in the ordinary way from good pig free from phosphorus is said to be by the *acid process*, while that made from pig iron containing considerable quantities of phosphorus in a furnace or converter lined with magnesian limestone gannister is called *basic steel*, and said to be made by the Thomas-Gilchrist or *basic process*. Basic steel is generally softer than ordinary steel; but excellent steel or ingot iron is made by this process, having an ultimate tensile strength of not more than 25 tons, with a stretch of over 30 per cent.; it can be bent cold two double, and is soft to work and easily welded. Large quantities of this material are now made at Middlesbrough, North Lincolnshire, &c., where there is an unlimited supply of suitable iron, and used by shipbuilders, tank and bridge builders, &c. The carbon and manganese are added as in the Bessemer process by the introduction of a small quantity of *ferro-manganese*.

At the present time steel plates, bars, and forgings are made almost exclusively from ingots run from either Bessemer converters or Siemens furnaces.

Puddled Steel.—Some years ago this material commanded considerable attention, since from it large shafts were made at what were then comparatively low prices, and it was thought that, from its superiority to wrought iron, it would take the place of the latter for forgings. To all intents and purposes it was only well worked "puddled" bar, containing more carbon than that used in making wrought iron, and was obtained by prematurely withdrawing the bloom before decarbonisation was completed.

The Steel used for Boiler Construction is made by the Siemens-Martin process exclusively. The ingots are reheated and hammered into slabs, which are again reheated and rolled into bars or plates. It is necessary that the material of which a boiler is constructed shall have very considerable elasticity as well as strength, and since best Yorkshire iron stretches to as much as 18 per cent. before fracture, steel should not be used which is not equal to this.

It is found that the lower the ultimate strength of good steel is, the higher is its elasticity, so that plates of this material, having an ultimate elongation of 20 per cent., possess a tensile strength of 32 tons per square inch while those having an ultimate strength of 26 tons will stretch 30 per cent. The former is very suitable for the shell-plating of boilers, as its strength is nearly 50 per cent. higher, and its elasticity nearly double that of the iron formerly used by boilermakers; the latter does eminently for the internal parts of a boiler which require to be of a somewhat softer material, that it may be flanged, &c., with ease, and stand the rough usage of the boiler smiths with safety. For this purpose plates are used which have a tensile strength of 26 to 28 tons per square inch, with an ultimate elongation of about 25 per cent. For corrugated furnaces a milder steel still is used, for its limit of strength is 25 tons, and it stretches to 30 per cent. in 8 inches Shell-plates are now made with a tensile strength of 35 tons, and a stretch of 25 per cent. with the grain and 20 per cent. across the grain.

Subjoined are the tests required by the various authorities for boiler steel. It will be seen that no two of them agree, which is to be much regretted; it is in the interest of both science and economy that they should all agree:—

Admiralty Tests of Material.—All steel to be made by the acid open hearth process. Every plate, &c., used is to be tested, and must comply with the requirements stated below:—

TABLE LIII.—Admiralty Tensile Tests.

Description of Material.	Minimum Ultimate Tensile Strength, Tons per square inch.	Maximum Ultimate Tensile Strength, Tons per square inch.	Minimum Elongation in 8 inches per cent.
			per cent.
Not exposed to flame and not flanged,	27	30	20
Exposed to flame and flanged, - -	24	27	25
Rivet bars, - - - - - -	24	27	25
Steam-pipe plates, - - - - -	24	27	33
Corrugated or ribbed furnace, - -	23	25	27
Tube forgings (annealed), - - -	21	24	27
Pieces cut from tubes (annealed), -	...	26	27

For bending tests the specimens are heated to a low cherry red, and then cooled in water at 82° F. Strips of plate 1½ inches wide must bend double in press, inner radius being 1½ times thickness of plate. For pieces of rivet bar inner radius to equal radius of bar; and for strips from tubes, ½ inch.

Board of Trade Tests of Material.—Strips 2 inches wide should be cut from at least one of every four ordinary plates, from each end of each plate over 15 feet in length, and from each corner of each plate over 20 feet × 6 feet, or over 2½ tons in weight; where more than one test piece is taken from a plate the *mean* result is to be adopted.

TABLE LIV.—Board of Trade Tensile Tests.

Description of Material.	Minimum Ultimate Tensile Strength, Tons per square inch.	Maximum Ultimate Tensile Strength, Tons per square inch.	Elongation in 10 inches per cent.
Plates not exposed to flame, - - - -	27	35	About 25 per cent., not less than 18 per cent.
Plates that are exposed to flame, - - -	26	30	Not less than 20 per cent. for annealed plates.
Rivet bars, - - -	26	30	Not less than 25 per cent.
Stay bars, - - -	27	32	About 25 per cent., not less than 20 per cent.
Tube strips, . -	26	30	About 25 per cent., not less than 20 per cent.
Rivets, - - -	27	32	Contraction of area about 60 per cent.

For bending tests the specimens should be 2 inches wide and 10 inches long, and should be bent double, the inner radius being 1½ times the thickness of the plate; for plates not exposed to flame, the tests should be made on the plate in its ordinary condition, and for plates that are exposed to flame the specimens should be heated to a cherry red, and cooled out in water at 80° F. before bending.

Lloyd's Tests of Material.—Ultimate tensile strength to be as follows:—Ordinary shell-plates and longitudinal stays, 28 to 32 tons per square inch, with a minimum elongation of 20 per cent. on standard test piece (8 inches or 8 diameters long); shell-plates, if to be flanged or welded, and all other plates, screw stays, and rivet bars, 26 to 30 tons. The minimum elongations on standard test pieces being as follows:—For shell-plates, if to be flanged or welded, all other plates, and screwed combustion chamber stays, 23 per cent.; for rivet bars, 25 per cent.

Test pieces, not less than 1½ inches wide, cut from plates or bars to stand bending to a curve, the inner radius of which is not greater than 1½ times thickness, both when in normal condition and after having been heated to a low cherry red and quenched in water at 80° F. Steel manufactured rivets to be capable of being bent cold and hammered until the shank is bent round on to itself without fracture on outside of bend. Heads to permit of being flattened hot to 2½ times diameter of shank.

One tensile, and either one cold or one temper bend test to be made from every plate; but for plates exceeding 2½ tons in weight, one tensile test from each end, and a cold bend test from one end and a temper bend test from the other are required. For each cast or furnace charge from which bars are rolled there must be two tensile tests, but when more than 15 bars are rolled from one charge, an additional tensile test is required for every further batch of 15 bars or portion thereof. One cold or one temper bend test is required from every bar as rolled, but no bend tests are required for rivet bars.

British Corporation Boiler Steel.—The quality of steel to be used in the construction of boilers must be as follows, viz. :—

Plates intended for the cylindrical shells and butt straps, and bars intended for stays, are to have an ultimate tensile strength of not less than 28 tons, or more than 32 tons per square inch, with at least 20 per cent. extension in 8 inches of prepared section. Plates which have to be flanged or welded, plates and stays for combustion-chambers, plates for furnaces, and bars intended for rivets are to be from 26 to 30 tons per square inch in tensile strength, with not less than 23 per cent. extension in 8 inches, but with rivet bars the extension is to be not less than 25 per cent. in 8 inches.

The sample pieces for temper bend tests are to be heated uniformly to a dull red, then cooled in water the temperature of which does not exceed 80° F., and the pieces in both cold and temper bend tests must withstand, without fracture, being bent double to a curve, the inner radius of which does not exceed 1½ times the thickness of the piece.

All plates which have been welded, locally heated, or in which rivet holes have been punched, and steel stay bars which have been worked in the fire must be subsequently annealed, and in no case are steel stays to be welded. Number of tests same as for Lloyd's, see above.

John Spencer & Sons, Newburn, David Colville & Sons, Motherwell, and others, manufacture plates and forgings from patent steels which have an ultimate tensile strength of 40 tons, with an elongation of 30 per cent. in 2 inches and a limit of elasticity of 25 tons. Such plates, however, cost a little more to manufacture than the ordinary Siemens steel.

Ordinary mild bar steel, such as used for the stays of boilers, for bolts, studs, &c., has a tensile strength of 26 to 32 tons, with an elongation of 25 to 20 per cent., and a harder steel for pins, &c., has 35 to 40 tons, with 15 per cent.

The specific gravity of mild steel plates and bars is about 7·86; the weight of a cubic foot is 491 lbs.; that of a cubic inch is 0·284 lb., and that of a square foot, 1 inch thick, 40·94 lbs.

Steel Forgings.—Shafts, piston- and connecting-rods, valve-rods, gudgeons, &c., are now generally made of steel forged from ingots manufactured by the Siemens process. The steel is, of course, of a mild kind, and while possessing properties very similar to those of the rolled bars and plates, is not quite so uniform in structure and strength.

There is little doubt that the ultimate strength of marine shafts, when made of steel, does not exceed 35 tons on the average, and those over 12 inches diameter cannot be depended on for a higher average than 30 tons, however well forged. The material near the centre of a steel forging of large size remains but very little affected by hammering, so that the larger the diameter of the shaft, the less will be the average strength of the material composing it. Forgings made with a hydraulic press are, however, free from this suspicion.

Small steel forgings may have as high a tensile strength as bars rolled from similar ingots, but as a rule they have not so great elasticity, and it is safer, therefore, to suppose them to be 10 per cent. weaker, for purposes of calculations, although, as a matter of fact, their tensile strength is sometimes higher.

Steel Castings.—Many parts of a marine engine which were formerly made of forged iron, are now made with advantage of cast steel; other parts which, for convenience of manufacture, were made of brass, are also made of this material. It is likely to supersede cast iron in many parts which must of necessity be cast, and as the cost of production of cast steel is reduced, so will the demand for it increase.

The chief obstacle to its further extended use is the uncertainty as to the soundness of the castings. Continued use, however, is fast dissipating the prejudice which once existed against its application, and as the demand for these castings has caused the manufacturers of them to give the closest attention to their production, the day is not far distant when a steel casting will command the same confidence as to its soundness that now obtains for iron castings. Indeed, an iron casting is more treacherous than one of steel, because blow-holes and spongy places are always near the surface of the latter, and can often be detected, while those in the iron castings are quite hidden. If a steel casting is machined, so that the faulty places are cut away, the part remaining may be depended on as quite sound.

Propeller blades and bosses, foundations, columns, levers, cross-heads for piston-rods, pistons of all sizes and shapes, link-motion blocks, eccentric straps, worms, wheels, &c., are now very generally made of cast steel; crank-shafts, too, have been made of cast steel, and did their work very satisfactorily, even when of so large a size as 15 inches diameter; connecting-rods, also, have been made of this material, and the economy of production of such parts when thus made is beyond all doubt. This material has the merit of possessing a strength at the centre as high as that at the circumference, whereas this is not always the case with large forgings.

The probable *average* strength of best steel castings is about 28 tons per square inch, with an elasticity of about 25 per cent.; greater strength is easily obtained with steel castings, but the elasticity will then be much lower, in fact, it is only by using very carefully selected materials, and by annealing after casting, that *so low* a strength can be obtained. The ordinary steel castings have an average ultimate strength of about 32 tons, with an elongation of about 18 per cent. when sound. To allow for unsoundness the ultimate strength may be assumed to be only 28 tons for the harder varieties, and 25 tons for the softer ones; for propeller blades it should not be assumed to be higher than 24 tons, although the sound parts of such castings are often found to have a strength of over 30 tons per square inch.

Nickel Steel.—This material, which is an alloy of steel and nickel, or, to be more precise, of iron, carbon, manganese and nickel, is likely to be generally used in the future where lightness is of first importance in engineering structures, or when the strongest possible material is required. Great care, however, has to be taken when forging and otherwise treating this material in the hot state to avoid seriously injuring its physical properties. It is, therefore, better to have the forging, &c., delivered from the maker's works ready for machining, and in all cases it should be most carefully annealed. It is also advisable to have it oil-tempered as well as annealed to get the full advantages from this very splendid alloy. Mr. Wm. Beardmore read a very interesting paper to the Institution of Naval Architects, in 1897, on the nickel-steel alloys his firm was making for boiler plates; with an ultimate strength of 74,000 lbs., and an extension of 23 per cent., the elastic limit was 43,000 lbs.; and a stronger quality whose ultimate strength was 114,000 lbs., with 25 per cent. extension, the elastic limit was 64,000 lbs. Forgings on a large scale have been made in this country by Messrs. Vickers, Sons & Maxim, Beardmore, and others for marine work as well as armour plates; while in America, at the Bethlehem Iron Works, this material is very largely used for marine work. Steel having 0·25 to 0·45 of carbon seems best for this alloy, and for marine purposes 3 to 5 per cent. of nickel is used with it. Less than 3 per cent. nickel is of no advantage.

Forgings having 0·29 per cent. of carbon and 3·25 of nickel have an ultimate strength of 85,000 lbs., with 30 per cent. elongation, and an elastic limit of 54,000 lbs.; with 3·75 per cent. of nickel and oil tempered, 90,000 lbs., with 22 per cent. extension, and 60,000 lbs. elastic limit; with 4·5 to 5 per cent. the elastic limit is as high as 80,000 lbs., and the extension 20 per cent. after careful oil tempering and annealing. The coefficient of expansion of an alloy containing 25 to 30 per cent. of nickel is exceedingly low, and its electrical resistance is 40 times that of copper; it is also nearly incorrodible. Adding 3 to 3½ per cent. of nickel increases the cost of a steel by about 7s. per cwt. This material stands percussion shocks exceedingly well, and also resists vibratory stresses equally well. Iron bars stressed to 40,000 lbs. stood only 59,000 alternations, mild steel 170,000, high carbon (0·35 per cent.) 317,000, while nickel steel (0·25 C, 3·25 Ni) stood 1,850,000, and 5½ per cent. nickel as many as 4,370,000 alternations.

Manganese Steel.—The invention of Mr. Robert Hadfield is of interest, although it is not yet generally useful to the engineer. That best known is an alloy containing 11 to 14 per cent. of manganese, and is largely used for the wearing parts of dredgers, crushing machinery, and ash ejectors. It is very tough and very hard, so that while it can be forged easily and bent readily when cold, it cannot be machined. In the forged condition it has a tenacity of 60 tons per square inch, with an elongation of 35 to 40 per cent. in 8 inches. At present articles are forged to shape or ground, and it is hoped that soon some method of machining this valuable material will be found.

Copper.—This metal is used to form alloys more frequently than by itself; it is too soft for general purposes; but as it is so much improved by the addition of even small quantities of other metals, it is perhaps, next to iron, the most important of the metals used by the marine engineer.

In commerce there are three distinct qualities of ingot copper dealt with, viz.:—(1) G.M.B. (good merchantable brand), a general term used as a basis for price; (2) *Tough Ingot*, which is generally used by founders for general purposes, and branded with some well-known smelter's name or mark; (3) *Best Selected*, is purer and of more uniform quality, and used for high class work.

In its simple state it is employed chiefly for pipes on account of its ductility and strength, and in some measure because it can be joined by brazing, so as to be as strong there as the original sheet. It does not generally corrode under the action of sea-water or air, but does sometimes waste by the mechanical action of water and steam moving at high velocities over its surface. In some few localities the water seems to have a destructive effect on this metal, owing no doubt to presence of free gases mechanically mixed with it; of these gases, sulphuretted hydrogen is the worst in the water of rivers and ports, and chlorine in sea-water; the presence of bromine, too, is often detected by the smell in the hot-wells of some engines.

Copper pipes of large size (6 inches and upwards) are always made from sheets, curved into the required form by rolling or hammering, and brazed at the seams. Copper pipes can be made seamless, practically of any size, by Elmore's Patent Copper Depositing Co., Leeds, by electro deposits, &c.

Smaller pipes are sometimes made in the same way, and sometimes by "drawing" in a way similar to that by which wire is manufactured. The feed, blow-off, and scum pipes in the Navy are always of solid-drawn copper, because there should be no seam in such important pipes; and the main steam pipes have also to be solid drawn when made of copper, and not exceeding 6 inches diameter.

Solid-drawn pipes are very seldom used in the mercantile marine, partly because they are somewhat more expensive, but chiefly because they are not so uniform in thickness as are the brazed ones, and are thought to be more liable to split unless carefully manufactured from very soft tough copper.

The strength of copper depends somewhat on its purity, but principally on the amount of work it has undergone. Copper castings have an ultimate strength of only about 10 tons per square inch; when forged its strength is increased to about 15 tons, and when rolled into bars is 16 tons; if a small proportion of phosphorus is added (about 2 per cent.) the strength is increased to 20 tons; pure copper when drawn out into wire has a strength of about 28 tons before, and 18 tons after annealing. Sheet copper has an average strength of about 13½ tons, and for purposes of calculation may be assumed to be 30,000 lbs. per square inch. Small quantities of arsenic and some other metals improve copper very much. With only ½ per cent. of arsenic plates have an ultimate strength of 33,420, with a stretch of 37 per cent (Roberts-Austen). Wire made of copper with 1 per cent. of aluminium stood 78,000 lbs. per square inch, and French wire made of copper with a small addition of silicon 128,000 lbs. Wire with 0·529 per cent. of antimony stood 78,000 lbs. as against 49,000 lbs. with pure copper wire.

The specific gravity of sheet copper is 8·805; the weight of a cubic foot is 550 lbs., that of a cubic inch 0·318 of a pound, and that of a square foot 1 inch thick 45·83 lbs.

Tin.—This metal, although seldom used alone, is very important, as forming one of the chief constituents of bronze or gun-metal and some white metals. The best qualities are obtained from Cornwall, and the chief supply of this metal is from that county; of late years, however, considerable quantities have been imported from the Dutch East Indies and from Australia, and although not so pure as the Cornish tin, the price of the latter has been very considerably reduced by the supply.

Tin is used as a protective covering to other metals on account of its immunity from the corrosive action of salts and acids. The Admiralty require all condenser tubes to be coated with tin when fitted in iron condensers; and this practice is also followed by some Mercantile Shipping Companies. Sheet tin, which is thin sheet iron coated with tin, is used for "liners" between "brasses," as well as for making oil feeders, lamps, cups, &c.

Tin mixed with small quantities of copper, antimony, &c., is used under the name of white metal to line and face bearings.

The tensile strength of tin is too low to admit of its being used alone in construction; its ultimate strength when cast is only 2·11 tons per square inch. Its specific gravity is about 7·3, consequently the weight of a cubic foot is 456 lbs., and that of a cubic inch 0·264 lb.

Zinc or Spelter.—This metal also is seldom used alone, but is very largely employed to alloy with copper to form brass. The best kinds are imported from the Continent, and the Silesian spelter is the best and that generally used in making brass for rolling into sheets or drawing into tubes, pipes, and rods.

In its simple state zinc is used by marine engineers to prevent corrosion in the boilers, condensers, and hot-wells. Cast blocks, or, better still, pieces of rolled bar or sheet of this metal are placed in metallic contact with the iron of the boiler in such places as have been found by experience to require protection. The purer the zinc is, the more perfect is the protection afforded; but unless there are exceptional circumstances affecting the feed-water, common zinc or even "hard spelter" (residuum from the galvanising bath) will form a sufficiently strong galvanic couple to prevent deterioration of the iron surfaces.

Zinc is also employed as a covering for iron, to protect it from the action of sea-water, &c., and being much cheaper than tin, and easily applied to the surface of the iron, is used on a far more extended scale than tin. Zinc is also used as the principal constituent of some kinds of white metal made for bushes working in water.

The tensile strength of zinc is even lower than that of tin, being only 1·336 tons per square inch when cast. Its specific gravity is 7·0, consequently the weight of a cubic foot is 437 lbs., and that of a cubic inch 0·253 lb.

Lead is nearly always used alone, and the purer it is, the more valuable for engineering purposes. In the mercantile marine the bilge piping is often made of lead, and the pipes for emptying and filling the ballast tanks are also sometimes of lead. It is used for these purposes because of its resisting the corrosive action of sea and bilge water, and being very ductile it can be easily bent to follow the curves and corners of the ship's bottom.

Sheet lead is used to protect the covering of boilers from wet, and also to cover engine-room floors when made of wood. It is employed, too, for jointing pipes, &c., when the flanges are rough and uneven.

It is sometimes used by moulders to give a good colour to common brass for ornamental purposes, and to cause it to be readily turned in the lathe; but even the smallest addition of this metal tends to reduce the strength of brass, and it should be therefore generally avoided.

The tensile strength of sheet lead is only 0·81 ton per square inch, and that of lead pipe is 1 ton. The specific gravity is 11·418, consequently the weight of a cubic foot is 712 lbs., of a cubic inch 0·412 lb., and of a square foot 1 inch thick, 59·3 lbs.

Aluminium.—This metal, formerly so rare and very expensive, is doubtless destined to become a very important one to marine engineers from its extreme lightness, and its power of resisting corrosive action. Its price when introduced (about £1000 per ton) quite precluded its use, even to form an alloy; but to-day, when it can be purchased at £130 per ton, engineers can use it freely to make bronze, which is as strong and elastic as mild steel. Aluminium, with the addition of 6 per cent of copper, is rolled into sheets and bars having a fair strength and incorrodable, and cost only 1s. 6d. per lb. Such sheets have an ultimate strength of 11 to 12 tons annealed, and 14 to 16 tons unannealed, as against 7 to 8 tons in the case of sheets of pure aluminium. Aluminium bronzes in the cast state are very porous, and the crystalline structure very coarse. A little aluminium improves zinc bronzes, but damages tin bronzes.

In the pure state when drawn into wire, it has a tensile strength of about 8 tons per square inch; by hammering cold, the strength is raised from about 7 tons (as cast) to about 12 tons. An alloy of 90 per cent. of copper and 10 per cent. of aluminium, when rolled, has an ultimate tensile strength of 32 to 40 tons per square inch.

The specific gravity of aluminium is only 2·56, consequently a cubic foot weighs 160 lbs., a cubic inch 0·092 lb., and a square foot 1 inch thick, 13·33 lbs.

Antimony is used in very small quantities, to harden other metals and alloys.

Alloys.—Strictly speaking, only alloys of copper and zinc can be called *brass*, but ordinary bronze, as made for bearings, liners, and bushes, is often called brass, and from these circumstances the liners of journals and pin-bearings are called "brasses." Alloys of copper and tin, or those of copper and tin together with zinc or other metal, are called *bronze*. Alloys of copper and zinc with manganese or other modifier are generally called *zinc bronzes*.

Brass.—The *yellow brass* used for ornamental castings is usually composed of two parts of copper and one part of zinc; when carefully made, castings of yellow brass have a tensile strength of 12 to 13 tons, but the ordinary yellow brass, as supplied by founders, has a strength of only 10 to 11 tons; it is fairly tough, but too soft for general purposes.

Muntz's Metal, composed of three parts of copper and two of zinc, can be rolled out into bars and sheets, so as to have an average tensile strength of 22 tons per square inch, and in some cases bars of this metal have a strength as high as 27 tons. It is very ductile, and can be forged when hot; it will stretch very considerably before fracture, and may be used for springs when hammered or cold rolled, and not annealed.

Its specific gravity is 8·2, consequently a cubic foot weighs 512 lbs., that of a cubic inch 0·296 lb., and that of a square foot 1 inch thick is 42·7 lbs.

Naval Brass.—By the addition of 1 per cent. of tin to Muntz's metal, it has the power of resisting the action of sea-water, while retaining all its other properties. An alloy of 62 copper, 37 zinc, and 1 tin is known as *naval* brass, because of its extensive use in the construction of naval composite ships, and for the bolts of the engine-fittings which are exposed to sea-water in warships. This metal can be forged hot, and bent cold two double; its strength is superior to the ordinary Muntz's metal, and some specimens rolled cold and unannealed have been proved to have an ultimate strength of nearly 40 tons per square inch. As usually supplied, it has an ultimate strength of 27 tons, with an elongation of 19 to 20 per cent. in 10 inches; the elastic limit is about 19 tons.

Brass Tube Metal.—The ordinary brass condenser-tubes are made of a composition containing 70 per cent. of copper and 30 per cent. of zinc, but the Admiralty require them to be composed of 70 per cent. of best selected copper, 29 per cent. of Silesian zinc, and 1 per cent. of tin, and all tubes supplied to the Navy have to undergo the test described on p. 232.

Boiler tubes for the Navy were made of a composition of 68 per cent. of best selected copper and 32 per cent. of zinc. The strength of the metal of tubes made with 70 per cent. of best selected copper and 30 per cent. Silesian zinc is as high as 36 tons per square inch.

Gun-metal or Bronze.—There is no particular mixture to which this name belongs, as it is applied promiscuously to any composition of copper and tin, or copper, tin, and zinc.

The best known composition, and one which has high strength, is fairly hard, and very tough, is that containing 90 per cent. of copper and 10 per cent. of tin. Its tensile strength, when carefully made, is 17 tons per square inch; its specific gravity is 8·66, consequently a cubic foot weighs 561 lbs., a cubic inch 0·325 lb., and a square foot 1 inch thick 46·8 lbs. To insure sound castings, however, it is necessary to add a small quantity of zinc.

A much harder metal is made by mixing 84 per cent. of copper with 16 per cent. of tin; its tensile strength is 16 tons, specific gravity 8·56, and the weight of a cubic foot is 534 lbs.

For heavy bearings, where hardness is of more importance than strength, although the metal must not by any means lack strength, a good metal is made by mixing 79 per cent. of copper with 21 per cent. of tin; its tensile strength is nearly 14 tons when carefully made, and the average is 13½ tons; the specific gravity is 8·73, and the weight of a cubic foot is 544 lbs.

Admiralty Bronze.—The mixture specified for propellers and all bronze castings, &c., of warships is 87 per cent. of copper, 8 per cent. of tin, and 5 per cent. of zinc. When carefully made, this metal has a tensile strength of 15 tons, and the average strength of castings is about 13½ tons, with a stretch of 7½ per cent. in 2 inches.

Phosphor Bronze.—This metal is composed of copper and tin, with a small proportion of phosphorus. It is harder than the ordinary bronze, very close grained, and of superior strength. The average ultimate strength is about 17 tons per square inch, while that of some specimens is as high as 22 tons. The singular thing about this metal is its high elastic limit, being very nearly that of the ultimate strength. Its resistance to shearing in the cast state is also exceptionally high. Great care is required in melting, and repeated meltings very much reduce its virtue. It may be rolled out into extremely thin sheets, or drawn into wire, when the average tensile strength is 56 tons per square inch. Phosphor bronze sheet is used for the valves of air-pumps (*vide* p. 249).

This metal is also used for bearings, brasses, propeller-blades and bosses, pump-rods, &c.

Manganese Bronze.—This metal is good zinc bronze improved by the addition of ferro-manganese. The manganese is said to deoxidise any copper oxides which may be mechanically mixed with the copper, so "rendering the metal more dense and homogeneous." The No. 1 quality, which is used for forgings and rolling into rods, plates, sheets, angles, &c., when cast in metal moulds has an ultimate strength of 24 tons, and an elastic limit of 14 tons per square inch.

Rolled rods, plates, &c., have, when mild, an ultimate strength of 28 tons, and an elongation of 40 per cent.; but when so required it can be made with an ultimate strength of 30 to 32 tons, an elastic limit of 15 to 17 tons, and an ultimate elongation of 15 to 20 per cent.

By cold rolling the strength can be raised to even 40 tons, but the elongation is then reduced to 10 per cent.

There are various other qualities of manganese bronze manufactured to suit the different requirements of engineers.

This metal can be rolled, drawn, and forged, is very tough as well as strong, and is, therefore, a very valuable one for the marine engineer.

It is largely used for propeller blades, as by employing it they can be made very much thinner than if of ordinary bronze, or even cast steel.

Stone's Bronze is a zinc and copper composition, possessing all the characteristics of manganese bronze, and is used very extensively for propeller blades and engine castings, where high tensile strength and great toughness is necessary.

Bull's Metal is another of the zinc bronzes having a high tensile strength with great toughness; in the rolled state it has resistance to compression as high as 100 tons per square inch.

Delta Metal.—The invention of Mr. Dick is a high-class bronze, somewhat similar to manganese bronze, but rather harder and with a higher tensile strength. Castings made from it have an ultimate strength of 34 tons, and resist torsion better than the other bronzes, which are, as a rule, disappointing in this respect.

Parson's White Brass.—This is a most valuable material for facing and lining bearings; the composition of No. 2 quality is tin, 68; zinc, 30·5; copper, 1; and lead, 0·5; its success as a bearing metal is most unqualified, and it does well in crank-pin brasses at moderate speeds.

Babbit's White Metal.—This was, for very many years, almost the only white metal used for bearings, and until late years was without a rival. It is composed of 10 parts of tin, 1 of copper, and 1 of antimony.

A very good white metal is made by mixing 6 parts of tin with 1 of copper, 6 parts of tin with 1 of antimony, and adding the two mixtures together.

Plumtine is a comparatively cheap and very efficient white metal, composed of 48·5 per cent. of lead, 40·5 of tin, 11 of antimony, and 0·5 of copper.

Magnolia is also a white metal consisting largely of lead, and is well known as a bearing metal.

Fenton's White Metal, which is used for stern bushes and the bushes of paddle-wheels, &c., is composed of 8 parts of zinc, 1·66 of tin, and 0·44 of copper; it is fairly tough and hard, and in sandy water resists wear exceedingly well.

Stone's White Bronze is also an excellent metal for bearings and crank-pin brasses, especially of heavy engines running at high speed.

German Silver.—An alloy of nickel and copper and zinc, known by this name, is used largely for instruments and for domestic implements, after being electro-plated. It is of yellowish-white appearance, fairly hard and tough, and capable of being worked cold.

Basileus' Metal is white, in appearance like silver, it takes a high polish, and resists the action of damp air, sea water, and weak acids. In the cast state its ultimate strength is 14½ tons, and its limit of elasticity 9 tons. When rolled into sheets it has an ultimate strength of 32 tons, and in wire 37½ tons, with an elongation of 26 per cent.

Richard's Plastic Metal.—This is a white metal, which melts at a moderately low temperature, and can be worked with a soldering-iron. It is very useful, therefore, for mending bearings and for coating damaged brasses; it also does for a filling metal, working well in bearings and guides.

TABLE LV.—Composition of Metals and Alloys.

Metal or Alloy.	Ultimate Tensile.	Extension.	Iron. Fe.	Nickel. Ni.	Carbon. C.	Manganese. Mn.	Silicon. Si.	Copper. Cu.	Tin. Sn.	Zinc. Zn.	Aluminium. Al.	Antimony. Sb.	Arsenic. As.	Lead. Pb.	Phosphorus. Ph.
		Per cent.													
Pig iron, good grey, . .	...	...	95·0	...	2·99	...	0·97	...	...	...	...	...	...	...	0·50
,, common grey, .	...	...	94	...	3·44	0·43	1·13	...	...	...	...	...	...	...	1·24
,, hæmatite, . .	...	...	95·7	...	3·27	0·37	0·67	...	...	...	...	...	...	...	0 28
Steel, mild Siemens, . .	58,000	25	99·1	...	0·15	0·55	0·02	...	...	...	...	...	...	...	0·05
,, high tensile Siemens, .	80,000	22	...	...	0·40	...	...	...	...	...	...	...	...	...	...
,, for tools, ordinary, .	124,000	5	...	...	1·00	...	...	...	...	...	...	...	...	...	...
Nickel steel, mild forging, .	85,000	30	95·5	3·25	·285	·687	·082	...	...	...	...	...	...	...	0·04
,, strong ,, .	198,000	12	88·0	11·39	0·20	...	...	...	...	...	...	...	...	...	...
Manganese steel (Hadfield),	130,000	35	...	...	...	11·0	...	...	...	...	...	...	...	...	...
Copper plates, strong, . .	33,400	37	...	...	...	...	...	99·5	...	...	...	...	0·50	...	...
,, wire, . . .	78,000	...	...	...	...	...	...	99·0	...	...	1·00	...	...	...	...
,, ,, . . .	128,000	...	...	...	...	...	trace	97·1	1·14	1·12	...	...	...	...	...
Muntz metal, rolled, . .	56,000	27	...	...	...	...	...	60·0	...	40·0	...	...	...	...	...
Naval brass, ,, . .	60,800	19	...	...	...	...	...	62·0	1·00	37·0	...	...	...	...	...
,, condenser tubes, .	...	...	...	...	...	...	...	70·0	1·00	29·0	...	...	...	...	...
,, ,, plates, .	...	...	...	...	...	...	...	62·0	1·00	37·0	...	...	...	...	...
Aluminium sheets, . .	34,400	3·7	...	...	...	...	...	6·00	...	...	94·0	...	...	...	...
,, bronze, cast, .	60,400	25·6	...	...	...	...	2·00	90·5	...	...	7·5	...	...	...	...
,, ,, ,, .	56,000	42·0	...	...	...	...	...	95·0	...	...	5·0	...	...	...	...
Phosphor bronze (wheel), cast,	39,200	...	...	...	...	...	...	82·2	12·95	...	...	...	...	4·28	0·52
Manganese ,, (Parsons), ,,	67,000	21·5	1·67	...	0·06	16·86	...	81·0	...	...	...	...	...	...	...

TABLE LV.—Composition of Metals and Alloys (*Continued*).

Metal or Alloy.	Ultimate Tensile.	Extension.	Iron. Fe.	Nickel. Ni.	Carbon. C.	Manganese. Mn.	Silicon. Si.	Copper. Cu.	Tin. Sn.	Zinc. Zn.	Aluminium. Al.	Antimony. Sb.	Arsenic. As.	Lead. Pb.	Phosphorus. Ph.
		Per cent.													
Stone's bronze, cast, . .	77,500	...	1·67	...	...	...	...	56·1	1·05	40·6	...	...	...	0·47	...
Bull's metal, rolled, . .	78,000	34	...	...	...	...	...	57·1	1·49	40·2	0·15	...	...	0·34	...
Delta ,, ,, . .	89,000	...	0·69	...	...	2·06	...	57·4	...	38·9	...	...	...	0·71	...
Roma ,, ,, . .	77,500	...	0·03	...	...	...	...	58·6	...	40·7	0·20	...	...	0·39	...
Aich's ,, ,, . .	57,300	25·6	1·50	...	...	...	...	59·8	0·80	37·9	...	...	...	...	...
Sterro ,, ,, . .	85,000	...	1·50	...	...	...	...	60·0	...	38·1	...	...	...	...	...
German silver (English), .	...	...	...	19·1	...	...	...	61·3	...	19·1	...	...	...	...	...
Argentan (Basileus), . .	...	...	...	26·0	...	...	...	52·0	...	22·0	...	...	...	...	...
Gunmetal, ...	38,000	...	...	...	...	...	...	91·0	9·0	...	...	...	...	...	...
Admiralty bronze, . .	30,000	...	...	...	...	...	...	87·0	8·0	5·0	...	...	...	...	...
,, ,, for fittings,	...	...	...	...	...	...	...	87·0	11·0	2·0	...	...	...	...	...
,, white metal, .	...	...	...	...	...	...	...	7·15	85·7	...	...	7·15	...	...	...
Babbit's white metal, . .	...	...	...	...	...	...	...	8·33	83·3	...	...	8·33	...	...	...
Parsons' white brass, No. 2,	...	...	...	...	...	...	...	1·00	68·0	30·5	...	...	...	0·50	...
Magnolia metal, . . .	...	...	...	...	...	...	...	...	4·60	...	...	13·2	...	82·0	...
Plumtine ,, ...	...	...	...	...	...	...	...	0·50	40·5	...	...	11·0	...	48·5	...
Fenton's white metal, . .	...	...	...	...	...	...	...	4·40	16·6	79·0	...	...	...	...	...
Bell metal, ...	...	...	...	...	...	...	...	76·5	23·5	...	...	...	...	...	...
Britannia metal, . . .	...	...	...	...	...	...	...	1·46	90·6	...	...	7·81	...	...	...
Solder, fine,. . . .	...	...	...	...	...	...	...	...	66·6	...	...	...	...	33·4	...
,, plumber's, . .	...	...	...	...	...	...	...	...	33·3	...	...	...	...	66·7	...

TABLE LV.—COMPOSITION OF METALS AND ALLOYS.

Metal or Alloy.	Ultimate Tensile.	Extension.	Iron. Fe.	Nickel. Ni.	Carbon. C.	Manganese. Mn.	Silicon. Si.	Copper. Cu.	Tin. Sn.	Zinc. Zn.	Aluminium. Al.	Antimony. Sb.	Arsenic. As.	Lead. Pb.	Phosphorus. Ph.
		Per cent.													
Pig iron, good grey, . .	...	...	95·0	...	2·99	...	0·97	...	...	...	...	...	...	...	0·50
„ common grey, .	...	...	94	...	3·44	0·43	1·13	...	...	...	...	...	...	...	1·24
„ hæmatite, . .	...	...	95·7	...	3·27	0·37	0·67	...	...	...	...	...	...	...	0 28
Steel, mild Siemens, . .	58,000	25	99·1	...	0·15	0·55	0·02	...	...	...	...	...	...	...	0·05
„ high tensile Siemens, .	80,000	22	...	...	0·40	...	...	...	...	...	...	...	...	...	...
„ for tools, ordinary, .	124,000	5	...	...	1·00	...	...	...	...	...	...	...	...	...	...
Nickel steel, mild forging, .	85,000	30	95·5	3·25	·285	·687	·082	...	...	...	...	...	...	...	0·04
„ strong „ .	198,000	12	88·0	11·39	0·20	...	...	...	...	...	...	...	...	...	...
Manganese steel (Hadfield),	130,000	35	...	...	...	11·0	...	...	...	...	...	...	...	...	...
Copper plates, strong, . .	33,400	37	...	...	...	...	...	99·5	...	...	...	...	0·50	...	...
„ wire, . . .	78,000	...	...	...	...	...	...	99·0	...	...	1·00	...	...	...	...
„ „ . . .	128,000	...	...	...	...	...	trace	97·1	1·14	1·12	...	...	...	...	...
Muntz metal, rolled, . .	56,000	27	...	...	...	...	...	60·0	...	40·0	...	...	...	...	...
Naval brass, „ . .	60,800	19	...	...	...	...	...	62·0	1·00	37·0	...	...	...	...	...
„ condenser tubes, .	...	...	...	...	...	...	...	70·0	1·00	29·0	...	...	...	...	...
„ „ plates, .	...	...	...	...	...	...	...	62·0	1·00	37·0	...	...	...	...	...
Aluminium sheets, . .	34,400	3·7	...	...	...	...	...	6·00	...	...	94·0	...	...	...	...
„ bronze, cast, .	60,400	25·6	...	...	...	...	2·00	90·5	...	...	7·5	...	...	...	...
„ „ „ .	56,000	42·0	...	...	...	...	...	95·0	...	...	5·0	...	...	...	...
Phosphor bronze (wheel), cast,	39,200	...	...	...	...	...	...	82·2	12·95	...	...	...	...	4·28	0·52
Manganese „ (Parsons), „	67,000	21·5	1·67	...	0·06	16·86	...	81·0	...	...	...	...	...	...	...

TABLE LV.—Composition of Metals and Alloys (*Continued*).

Metal or Alloy.	Ultimate Tensile.	Extension. Per cent.	Iron. Fe.	Nickel. Ni.	Carbon. C.	Manganese. Mn.	Silicon. Si.	Copper. Cu.	Tin. Sn.	Zinc. Zn.	Aluminium. Al.	Antimony. Sb.	Arsenic. As.	Lead. Pb.	Phosphorus. Ph.
Stone's bronze, cast, . .	77,500	...	1·67	...	...	...	...	56·1	1·05	40·6	...	...	...	0·47	...
Bull's metal, rolled, . .	78,000	34	...	...	...	...	...	57·1	1·49	40·2	0·15	...	...	0·34	...
Delta ,, ,, . .	89,000	...	0·69	...	...	2·06	...	57·4	...	38·9	...	...	...	0·71	...
Roma ,, ,, . .	77,500	...	0·03	...	...	...	...	58·6	..	40·7	0·20	...	...	0·39	...
Aich's ,, ,, . .	57,300	25·6	1·50	...	...	...	...	59·8	0·80	37·9	...	...	...	...	...
Sterro ,, ,, . .	85,000	...	1·50	...	...	...	...	60·0	...	38·1	...	...	...	...	...
German silver (English), .	...	...	...	19·1	...	...	...	61·3	...	19·1	...	...	...	...	...
Argentan (Basileus), . .	...	...	...	26·0	...	...	...	52·0	...	22·0	...	...	...	...	...
Gunmetal, ...	38,000	...	...	...	...	...	...	91·0	9·0	...	...	...	...	...	...
Admiralty bronze, . .	30,000	...	...	...	...	...	...	87·0	8·0	5·0	...	...	...	...	...
,, ,, for fittings,	...	...	...	...	...	...	...	87·0	11·0	2·0	...	...	..	...	...
,, white metal, .	...	...	...	...	...	...	...	7·15	85·7	...	...	7·15	...	..	...
Babbit's white metal, . .	...	...	...	...	...	...	...	8·33	83·3	...	...	8·33	...	...	...
Parsons' white brass, No. 2,	...	...	...	...	...	...	...	1·00	68·0	30·5	...	...	...	0·50	...
Magnolia metal, ...	...	...	...	...	...	...	...	...	4·60	...	...	13·2	...	82·0	...
Plumtine ,, ...	...	...	...	...	...	...	...	0·50	40·5	...	...	11·0	...	48·5	...
Fenton's white metal, . .	...	...	...	...	...	...	...	4·40	16·6	79·0	...	...	...	...	...
Bell metal, ...	...	...	...	...	...	...	...	76·5	23·5	...	...	...	...	...	...
Britannia metal, ...	...	...	...	...	...	...	...	1·46	90·6	...	...	7·81	...	...	...
Solder, fine, ...	...	...	...	...	...	...	...	...	66·6	...	...	...	...	33·4	...
,, plumber's, . .	...	...	...	...	...	...	...	...	33·3	...	...	...	...	66·7	...

TABLE LV*a*.—COMPOSITION OF WHITE (BEARING) METALS.

Name of Metal.	Cu.	Sn.	Sb.	Zn.	Pb.	Fe.	Miscellaneous.
Dewrance's Locomotive,	22·2	33·3	44·4	...	...	...	...
For gland packings,	7·8	88·1	3·5	...	2·6	...	...
Used in the German navy,	7·5	85·0	7·5	...	...	...	...
,, French ,,	7·0	7·5	...	78·5	7·0	...	...
,, British ,,	5·5	86·0	8·5	...	...	...	...
Fenton's,	4·4	16·6	...	79·0	...	...	...
Magnolia,	...	...	21·0	...	78·0	1·0	...
,,	...	4·6	13·0	...	82·0	...	...
Kingston's,	6·0	88·0	...	...	...	...	6 Hg
Parson's white brass,	5·6	17·5	0·8	76·1	...	...	...
,, ,, metal,	...	58·5	2·0	39·5	...	...	...
,, ,, ,,	1·0	68·0	...	30·5	0·5	...	...
For common bearings,	10·0	...	10·0	...	80·0	...	...
,, heavily loaded bearings,	64·0	5·0	...	...	30·0	...	1 Ni
Plumtine,	0·5	40·5	11·0	...	48·5	...	...

TABLE LVI.—PROPERTIES OF METALS.

Metal or Alloy.	Ultimate Strength.	Extension.	Elastic Limit.	Compressed Strength.	Shear.	Specific Gravity.	Weight of		Melts at ° F.	Conducts Heat.	Extension per 10° F.
							Cubic Inch.	Sq. foot 1 inch Thick.		Silver 100	
	Lbs.	Per cent.	Lbs.	Lbs.	Per cent.						
Cast iron, good castings,	25,000	...	...	110,000	162·0	7·20	0·260	37·50	2100	...	...
Wrought iron, good boiler quality,	50,000	...	26,000	...	80·0	7·68	0·280	40·0	2800	11·90	·0000670
,, ,, bar ,,	60,000	...	30,000	...	...	7·62	0·276	39·70	,,	,,	,,
,, ship plate ,,	45,000	...	27,500	...	...	7·68	0·280	40·0	,,	,,	,,
Steel castings, good mild,	63,000	...	34,000	...	...	...	...	...	2500	,,	,,
,, wrought (0·15 carbon),	58,000	25·0	29,000	...	80·0	7·86	0·284	40·94	,,	,,	,,
,, ,, (0·40 ,,),	78,000	22·0	33,000	...	...	...	...	...	...	...	...
,, ,, (,, ,,) oil tempered,	90,000	...	55,000	...	...	...	...	...	...	...	...
Nickel steel (3¼ Ni, 0·29 C), forged,	85,000	30·0	54,000	...	...	...	...	...	...	...	...
,, (3¾ Ni), oil tempered,	90,000	...	60,000	...	...	...	...	...	...	...	...
,, mild quality boiler plate,	74,000	...	43,000	...	...	...	...	...	...	...	...
,, high tensile ,,	114,000	...	64,000	...	...	...	...	...	...	...	...
Chrome steel (2·5 Cr, 0·39 C), forged,	65,400	...	52,600	...	...	...	...	...	...	...	...
,, (,,), unannealed,	120,900	...	97,400	...	...	...	...	...	...	...	...
Manganese steel (11·0 Mn), forged,	134,000	35·0	...	...	...	...	...	...	...	...	...
Nickel, cast,	36,400	...	24,600	...	...	8·30	...	...	2820	...	...
,, forged,	72,100	...	25,500	...	...	8·80	0·318	45·80	2820	...	·0000705
German silver (19 Ni), rolled,	...	...	...	...	...	8·80	0·318	45·80	...	...	...
Copper, annealed sheets,	30,000	...	5,600	...	86·7	8·85	0·320	46·10	2000	73·6	·0000930
,, bars,	36,000	...	...	...	...	,,	...	...	,,	,,	,,
Zinc, cast,	3,000	...	...	...	...	7·00	0·253	36·42	800	28·1	·0001620
,, rolled,	7,500	...	...	...	...	,,	...	...	,,	,,	,,
Tin, cast,	4,700	...	...	...	...	7·30	0·264	38·00	440	15·2	·0001280
Muntz metal, rolled,	56,000	27·0	36,000	...	...	8·20	0·296	42·72	1630	...	...
Naval brass, ,,	60,800	19·0	42,400	...	55·8	8·20	0·296	42·72	1640	...	...

TABLE LVI.—PROPERTIES OF METALS (*Continued*).

Metal or Alloy.	Ultimate Strength.	Extension.	Elastic Limit.	Compressed Strength.	Sheer.	Specific Gravity.	Weight of		Melts at ° F.	Conducts Heat.	Extension per 10° F.
							Cubic Inch.	Sq. foot 1 inch Thick.			
	Lbs.	Per cent.	Lbs.	Lbs.	Per cent.						
Manganese bronze (Parsons), cast No. 2, .	67,000	24·5	33,000	...	...	8·51	0·308	44·34	1660	...	...
,, ,, rolled, annealed, ...	67,500	26·2	33,000	...	...	8·40	0·304	43·76	,,	...	...
Stone's bronze, cast, ...	77,500	...	38,200	127,000	...	...	...	...	...	...	...
,, rolled, ...	80,600	...	66,000	...	...	...	...	...	...	...	...
Delta metal, rolled, ...	89,000	...	...	...	...	8·40	0·304	43·76	1800	...	...
Bull's metal, cast, ...	70,000	...	29,600	176,000	...	8·30	0·300	43·24	1800	...	...
,, rolled, ...	78,000	34·0	60,000	224,000	...	,,	,,	,,	,,	...	...
Aich's metal, ,, ...	57,300	25·6	...	...	...	...	...	...	...	...	...
Phosphor bronze (wheel B), cast, . .	39,200	...	38,900	...	113·0	8·60	0·311	44·80	1900	...	...
,, ,, rolled, . .	68,500	...	62,000	...	63·0	,,	,,	,,	,,	...	...
Silicium bronze wire, ...	105,000	...	...	...	...	...	...	...	...	...	...
Admiralty bronze, cast, ...	30,000	...	17,000	...	...	8·66	0·325	46·80	1700	...	...
Aluminium (pure comml.), cast, . .	11,800	...	...	...	...	2·56	0·092	13·33	1160	31·33	·0000280
,, (,,), bar rolled, .	28,200	...	19,300	...	...	2·70	0·098	14·06	,,	,,	,,
,, (,,), plates, rolled, .	16,500	...	9,600	...	...	2·70	0·098	14·06	,,	,,	,,
,, (6·0 Cu), plates rolled, . .	34,400	3·7	30,600	...	...	3·07	0·111	16·00	...	...	...
Aluminium bronze (10 Al), rolled, . .	100,000	22·0	...	...	...	7·65	0·277	39·86	1750	...	...
,, (5 Al), cast, . .	56,000	50·0	...	...	...	...	...	...	...	...	...
Antimony, cast, ...	...	...	...	...	...	6·71	0·243	35·00	1166	4·03	...
Bismuth, ,, ...	...	...	...	...	...	9·80	0·355	51·05	514	1·80	...
Lead, sheet, ...	3,300	...	...	...	...	11·37	0·411	59·24	620	8·50	...
Manganese, cast, ...	...	...	...	...	...	8·01	0·289	41·52	3450	...	...
Platinum, rolled, ...	...	...	...	...	...	21·50	0·778	112·0	3225	8·40	...
Silver, rolled, ...	...	...	..	...	...	10·53	0·381	54·86	1730	100·0	...

TABLE LVI*a*.—Prices of Materials.

Materials at Maker's Works.	Prices, 1908.			Prices, 1906.			
Pig iron, Blaenavon, cold blast, No. 3,	£8	0	0	£9	2	6	per ton.
,, Shropshire, ,, ,,	5	0	0	5	12	6	,,
,, Hæmatite, West Coast, ,,	2	18	0	3	5	0	,,
,, ,, East Coast, ,,	2	17	0	3	8	6	,,
,, Scotch, good brand, ,,	2	13	0	3	2	6	,,
,, ,, ,, No. 1,	£3 3s. to 3	7	0	3	8	0	,,
,, Cleveland, ,, No. 3,	2	7	0	2	9	0	,,
,, ,, ,, No. 1,	2	9	0	2	10	0	,,
,, Lincoln, ,, Foundry,	2	9	6	2	18	0	,,
,, Derby, ,, ,,	2	11	0	3	1	6	,,
Wrought Iron, Cleveland bars, ordinary,	6	10	0	7	5	0	,,
,, ,, ,, best,	6	15	0	7	15	0	,,
,, ,, ship plates,	6	15	0	7	5	0	,,
,, ,, ,, angles,	6	7	6	7	5	0	,,
,, Staffordshire, best bar,	7	0	0	8	0	0	,,
,, sheets,	8	15	0	8	5	0	,,
Steel, angles and tees, ship quality,	5	10	0	6	12	6	,,
,, plates, ship quality,	5	15	0	7	0	0	,,
,, boiler plates, Board of Trade, &c.,	7	5	0	8	0	0	,,
,, forgings, plain shafts, with couplings,	£10 to 15	0	0	£14 to 18	0	0	,,
,, for tools, common,	4d. to 6d.			......			per lb.
,, ,, good,	6d. to 9d.			......			,,
,, ,, self-hardening, for high speed,	£0	1	3	......			,,
,, ,, very best, for high speed,	0	2	0	......			,,
Copper ingots, G.M.B.,	53	10	0	79	5	0	per ton.
,, ,, tough,	56	0	0	84	5	0	,,
,, ,, best selected,	57	0	0	84	10	0	,,
,, locomotive plates,	72	0	0	(basis) 94	10	0	,,
,, sheets,	68	0	0	,, 92	0	0	,,
,, pipes, solid drawn, 1¼ in. to 3 in.,	0	0	8½	,, 0	0	10¾	per lb.
,, ,, ,, under 1¼ in. and over 3 in.,	0	0	8¾	,, ...			,,
,, ,, brazed,	0	0	8½	,, 0	0	10½	,,
Brass condenser plates, Mercantile,	0	0	6	0	0	7⅞	,,
,, ,, ,, Naval,	0	0	6¼	0	0	8⅛	,,
,, ,, tubes 66 per cent. copper,	0	0	7⅞	0	0	9	,,
,, ,, ,, 70 ,, ,, Mercantile,	0	0	8	0	0	9⅛	,,
,, ,, ,, 70 ,, ,, Admiralty,	0	0	8¼	0	0	9½	,,
Tin, English ingots,	124	0	0	166	0	0	per ton.
Zinc, slabs, Silesian,	19	10	0	25	2	6	,,
,, ,, best,	19	15	0	25	12	6	,,
Aluminium, ingots, Nos. 1, 2, and 4,	130	0	0	160	0	0	,,
,, ,, No. 6,	120	0	0	150	0	0	,,
,, sheets, from Nos. 1 to 30, wire gauge,	£154 to 224	0	0	£150 to 180	0	0	,,
Lead, pig,	11	5	0	17	15	0	,,
,, pipe,	£14 10s. to 15	0	0	19	10	0	,,
Nickel ingots, of good quality,	0	1	0	0	1	0	per lb.
,, refined (98 to 99 per cent. purity),	0	1	8	......			,,
German silver sheet,	0	1	2	1s. 2d. to 0	1	5	,,
Muntz metal sheet,	0	0	6	0	0	7¾	,,
Naval brass rods,	0	0	6¼	0	0	8⅛	,,
Phosphor bronze, ingots, for general castings,	87	10	0	102	0	0	per ton.
,, ,, suitable for bearings,	92	0	0	113	0	0	,,
,, rods,	0	0	11½	(basis) 0	1	2½	per lb.
,, sheet,	0	0	11	,, 0	1	4	,,
Manganese bronze, ingots,	£70 to 80	0	0	£90 to 105	0	0	per ton.
Stone's bronze, ingots for ordinary castings,	70	0	0	......			,,
Antimony, cakes,	30	0	0	68	0	0	,,
Bismuth,	0	5	6	0	7	0	per lb.
Platinum wire,	3	19	0	4	11	6	per oz. Troy.
,, sheets,	4	5	0	4	11	6	,,

TABLE LVII.—Standard Tests of Steel as Adopted in America.

	Specification for	Tensile Strength.	Yield Point.	Elongation (c).	Contraction of Area.	Bending Test. D = Diameter. T = Thickness.
		Lbs. per Square Inch.	Lbs. per Square Inch.	Per cent.		
Steel castings,	Ordinary castings,	...	...	...	...	...
	Tested castings, hard,	85,000	38,250	15	20	...
	,, medium,	70,000	31,500	18	25	90°, D = 2 T.
	,, soft,	60,000	27,000	22	30	120°, D = 2 T.
,, axles,	Car, engine truck, and tender truck,	...	...	...	...	...
	Driving axles (carbon steel),	80,000	40,000	18	...	...
	,, (nickel ,,),	80,000	50,000	25	45	...
,, forgings (a),	Soft or low carbon steel,	58,000	29,000	28	35	180°, D = T.
	Carbon steel, not annealed,	75,000	37,500	18	30	180°, D = 3 T.
,, tyres,	Passenger engines,	100,000	...	12	...	...
	Freight engines and car wheels,	110,000	...	10	...	...
	Switching engines,	120,000	...	8	...	...
Structural steel for buildings,	Rivet steel,	50,000 to 60,000	30,000	26	...	180°, flat.
	Medium steel,	60,000 ,, 70,000	35,000	22 (d)	...	180°, D = T.
,, ,, bridges and ships,	Rivet steel,	50,000 ,, 60,000	30,000	26	...	180°, flat.
	Soft steel,	52,000 ,, 62,000	32,000	25	...	180°, ,,
	Medium steel,	60,000 ,, 70,000	35,000	22 (d)	...	180°, D = T.
Open hearth boiler plate and rivet steel,	Flange or boiler steel,	55,000 ,, 65,000	33,000	25	...	180°, flat.
	Fire-box steel,	52,000 ,, 62,000	32,000	26	...	180°, ,,
	Extra soft steel,	45,000 ,, 55,000	30,000	28 (d)	...	180°, ,, (e)
Steel rails (b),		...	...	...	...	...
,, splice bars,		54,000 to 64,000	32,000	25	...	180°, flat.

(a) In the four classes of steel forgings not tabulated above, it will be found, by reference to the specifications, that elastic limit is specified instead of yield point.

(b) This table does not include the drop test required on castings, axles, tyres, and rails; the homogeneity test for fire-box steel; nor the percussive test for large steel castings.

(c) The elongation given for castings, axles, forgings, and tyres are on 2-inch the others on 8-inch gauged length.

(d) Reference to the text of the specifications must also be made for the variation in elongation allowed for thin and thick structural steels and boiler steel, and for the tensile tests of full-sized eye-bars.

(e) These same bending tests are required after quenching. (See Specification for details.)

TABLE LVIII.—Steel Forgings (American Standard).

Process of Manufacture.

1. Steel for forgings may be made by the open hearth, crucible, or Bessemer process.

Chemical Properties.

2. There will be four classes of steel forgings, which shall conform to the following limits in chemical composition :—

	Forgings of Soft or Low Carbon Steel.	Forgings of Carbon Steel not Annealed.	Forgings of Carbon Steel, Oil Tempered or Annealed.	Forgings of Nickel Steel, Oil Tempered or Annealed.
	Per cent.	Per cent.	Per cent.	Per cent.
Phosphorus shall not exceed	0·10	0·06	0·04	0·04
Sulphur " "	0·10	0·06	0·04	0·04
Nickel " "	...	...	...	3·75

Physical Properties.

3. The minimum physical qualities required of the different sized forgings of each class shall be as follows :—

Tensile Tests.				
Tensile Strength.	Yield Point.	Elongation in 2 Inches.	Contraction of Area.	
Lbs. per Sq. Inch.	Lbs. per Sq. Inch.	Per cent.	Per cent.	*Soft Steel or Low Carbon Steel.*
58,000	29,000	28	35	For solid or hollow forgings, no diameter or thickness of section to exceed 10 inches.
				Carbon Steel, not Annealed.
75,000	37,500	18	30	For solid or hollow forgings, no diameter or thickness of section to exceed 10 inches.
	Elastic Limit.			*Carbon Steel, Annealed.*
80,000	40,000	22	35	For solid or hollow forgings, no diameter or thickness of section to exceed 10 inches.
75,000	37,500	23	35	For solid forgings, no diameter to exceed 20 inches, or thickness of section 15 inches.
70,000	35,000	24	30	For solid forgings, over 20 inches diameter.
	Yield Point.			*Carbon Steel, Oil Tempered.*
90,000	55,000	20	45	For solid or hollow forgings, no diameter or thickness of section to exceed 3 inches.
85,000	50,000	22	45	For solid forgings of rectangular sections not exceeding 6 inches in thickness, or hollow forgings, the walls of which do not exceed 6 inches in thickness.

TABLE LVIII.—Steel Forgings (*Continued*).

Tensile Tests.				
Tensile Strength	Yield Point.	Elongation in 2 Inches.	Contraction of Area.	
Lbs. per Sq. Inch.	Lbs. per Sq. Inch.	Per cent.	Per cent.	*Carbon Steel, Oil Tempered—Continued.*
80,000	45,000	23	40	For solid forgings of rectangular sections not exceeding 10 inches in thickness, or hollow forgings, the walls of which do not exceed 10 inches in thickness.
				Nickel Steel, Annealed.
80,000	50,000	25	45	For solid or hollow forgings, no diameter or thickness of section to exceed 10 inches.
80,000	45,000	25	45	For solid forgings, no diameter to exceed 20 inches, or thickness of section 15 inches.
80,000	45,000	24	40	For solid forgings, over 20 inches diameter.
				Nickel Steel, Oil Tempered.
95,000	65,000	21	50	For solid or hollow forgings, no diameter or thickness of section to exceed 3 inches.
90,000	60,000	22	50	For solid forgings of rectangular sections not exceeding 6 inches in thickness, or hollow forgings, the walls of which do not exceed 6 inches in thickness.
85,000	55,000	24	45	For solid forgings of rectangular sections not exceeding 10 inches in thickness, or hollow forgings, the walls of which do not exceed 10 inches in thickness.

4. A specimen, 1 inch by ½ inch (1" × ½"), shall bend cold at 180° without fracture on outside of bent portion, as follows:—

Bending Test.	
Around a diameter of ½ inch,	For forgings of soft steel.
½ ,,	,, carbon steel, not annealed.
1½ ,,	,, ,, annealed, if 20 inches in diameter or over.
1 ,,	,, ,, ,, if under 20 inches diameter.
1 ,,	,, ,, oil tempered.
½ ,,	,, nickel steel, annealed.
1 ,,	,, ,, oil tempered.

Solid Matter in Solution in Water.—Professor Vivian Lewes gives the following analyses as fairly representing the salts, &c., held in solution in (1) ordinary river water, (2) in tidal rivers near their estuaries and well water near the sea, and (3) in ordinary sea water :—

TABLE LIX.

Constituents.	River.	Brackish.	Sea.
Calcic carbonate, ...	75·85	43·65	0·97
,, sulphate, ...	3·68	34·78	85·53
Magnesic hydrate, ...	2·56	4·34	3·39
Sodic chloride, ...	0·45	0·56	2·79
Silica, ...	7·66	7·52	1·10
Oxides of iron and alumina, ...	2·96	3·44	0·32
Organic matter, ...	3·64	1·55	trace
Moisture, ...	3·20	4·16	5·90
	100·00	100·00	100·00

The following are given by him as analyses of the deposits found in a marine boiler using salt water as its extra supply and the engine getting its internal lubrication in the usual way with *valvoline*, which is a mineral oil having a specific gravity of 0·889 and the boiling point 700° Fah. It is quite free from free acids, and should be quite free from any animal or vegetable oil or fat, and therefore highly suitable for such a purpose.

TABLE LX.—Boiler Deposits.

Scale from Furnace.			Deposit from Tubes.		
	From Top.	From Below.		Scale on Tubes.	Deposit above Scale.
Calcic sulphate, . .	84·87	59·11	Calcic sulphate, . .	50·92	11·60
,, carbonate, . .	5·90	6·07	,, carbonate, . .	4·18	0·82
Magnesic hydrate, .	2·83	11·29	Magnesic hydrate, .	14·12	22·21
Iron, alumina, and silica,	2·37	2·85	Iron, alumina, silica, &c.,	7·47	9·14
Organic matter and oil,	3·23	19·54	Organic matter and oil,	21·06	50·20
Moisture, . . .	0·80	1·14	Moisture, . . .	1·17	4·23
Alkalies, . . .	nil	nil	Alkalies, . . .	1·08	1·80
	100·00	100·00		100·00	100·00

Deposit from Bottom of Boiler.			
Calcic sulphate, ...	22·52	Moisture, ...	5·79
,, carbonate, ...	nil	Alkalies, ...	1·80
Magnesic hydrate, ...	7·09		
Silica, alumina, and iron, . .	34·85		
Organic matter and oil, . .	27·95		100·00

On careful examination of the organic matter and oil present in these deposits, it was found that quite one-half of it was "valvoline," in an unchanged condition, which had collected round small particles of calcic sulphate.

The Effect of Heat on the Strength of Metals.—In these days of high pressures, and with superheating likely to become general, it is of very great importance to know how far the metals used by engineers are affected by rise of temperature both in strength and structure. Steam (saturated) at a pressure of 150 lbs. above the atmosphere has a temperature of 365° Fah.; at a pressure of 200 lbs. the temperature is 387° Fah.; at 250 lbs., 406° Fah., and at 300 lbs. 421° Fah. Superheating may raise the temperature almost to any degree, and on shore as much as 600° Fah. is spoken of as desirable. In any case, however, *within* the H.P. cylinder the temperature will seldom much exceed that of the saturated steam of the same pressure as in the valve-box. It may, therefore, be taken as a general rule that the H.P. cylinder and its internal fittings must be suitable for a temperature of 450° Fah.; the medium-pressure cylinder for 350° Fah., and the low-pressure cylinder for 250° Fah., and the condenser top, 200° Fah.

Cast Iron is little affected by such temperatures as the above. Tredgold states that cast iron was by no means reduced in strength by heat up to 600° Fah.

Wrought Iron and Steel (Mild Ordinary) gain in ultimate strength with rise of temperature, so that at 400° Fah. it is 10 to 15 per cent. higher than when at 60° Fah., but with a reduction in the extension.

Copper.—Pure copper is seldom or never used in modern engineering. Commercially, pure copper, as in pipes and sheets, is not much affected by heat up to 500° Fah. Sir W. Roberts-Austen found that very pure copper having a tensile strength of 29,600 lbs., with 27 per cent. extension at 59 Fah., had 30,900 lbs. with 30 per cent. at 212°; 30,090 lbs. with 23 per cent. at 304°; 29,020 lbs. with 25 per cent. at 424°; 28,240 lbs. with 37 per cent. at 480°; and 25,390 lbs. with 21 per cent. at 617°. Also that copper containing only 0·2 per cent. of arsenic had an ultimate strength of 30,150 lbs. with 30 per cent. extension at 64° Fah., and 30,880 lbs. with 30 per cent. at 214°; 30,960 lbs. with 21 per cent. at 290°, and 28,730 lbs. with 30 per cent. at 480°. Such copper as this is very suitable for steam pipes. Copper containing 0·5 per cent. of arsenic is also good for locomotive fire-box plates as at 835° Fah., and its tensile strength as much as 20,000 lbs., with 13 per cent. extension. Professor Unwin states that rolled copper, which, at 60° Fah., had a strength of 40,000 lbs., had 38,500 at 212°, 36,000 at 350°, and 33,000 at 536°.

Gunmetal or Admiralty Bronze, which at 60° may be taken as having a tensile strength of 15·3 tons with an extension of 12·5 per cent., has 31,000 lbs. with 10 per cent. at 200°; 29,800 lbs. with 8·25 at 350°; but only 15,700 lbs. with 0·75 per cent. at 400°; at 500° Fah. the strength is about the same, but with no extension whatever. Gunmetal with *more zinc* in its composition, say, 15 per cent., has a tensile strength of 29,000 lbs. with an extension of 26 per cent.; at 400° Fah. it is still 27,750 lbs. with 25 per cent., but at 450° it drops to 9700 lbs. with 1·2 per cent. only.

Naval Brass, which at 60° has a strength of 60,000 lbs. with an extension of 19 per cent., still has 52,000 lbs. with 16 per cent. at 400°; at 500° the strength is about the same, with an extension of over 13 per cent.

Phosphor Bronze (cast) with a tensile strength of 39,000 lbs. and an extension of 17·5 per cent. at 60° Fah. has about the same strength at 200°; 37,000 lbs. with 15 per cent. extension at 250°; 36,700 with 12 per cent. at 300°; 29,400 with 7 per cent. at 350°; and 27,000 with 5 per cent. at 400° Fah., and the same at 500° Fah. Rolled phosphor bronze,

TABLE LXI.—EFFECT OF HEAT ON METALS.

Metal or Alloy.	At 100° F.		At 250° F.		At 350° F.		At 400° F.		At 450° F.		Melting Point. F.
	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	
		Per cent.		Per cent.		Per cent.		Per cent.		Per cent.	
Copper (0·2 per cent. arsenic), .	30,325	30·3	31,000	26·2	30,950	21·1	29,750	27·0	29,000	29·0	2,000°
Admiralty bronze, cast, . . .	32,000	12·0	30,800	9·3	29,800	8·25	15,700	0·75	15,500	0·2	1,700°
Mercantile ,, (soft), cast, .	29,000	26·0	28,800	25·8	28,000	23·0	27,750	25·0	9,700	1·2	1,680°
Naval brass, rolled, . . .	58,700	18·8	54,500	17·6	52,600	16·6	52,000	16·0	52,000	14·0	1,640°
Phosphor bronze, wheel brand, cast,	39,000	17·8	37,000	15·0	29,400	7·0	27,000	5·0	27,000	5·0	1,900°
,, ,, rolled,	68,250	6·8	68,000	7·0	67,750	7·5	64,500	8·1	60,000	9·0	1,900°
Manganese bronze, Parsons' No. 6, cast, ...	76,000	25·0	...	...	...	...	...	...	56,000	27·0	1,660°
Aich's metal, rolled, . . .	55,250	...	47,000	...	42,300	...	39,750	...	37,000	...	1,650°
Bull's metal, ,, . . .	70,000	16·0	63,600	13·6	58,700	13·0	56,750	14·6	...	...	1,800°
Aluminium, ...	25,300	...	20,250	...	15,500	...	13,300	...	11,500	...	1,160°

with a strength of 68,500 lbs., and an extension of 6·3 per cent. and an *elastic limit* of 62,000 lbs. has 67,500 lbs. with 8·2 per cent., and an elastic limit of 64,500 lbs. at 400° Fah.; and at 500° Fah. the strength is as high as 60,000 lbs. with 9·0 per cent. extension, the elastic limit falling to 54,000 lbs. only.

Bull's Metal, of soft quality, rolled, has a strength of 73,000 lbs. with an extension of 16·4 per cent., the elastic limit being 64,000 lbs.; at 300° Fah. it stood 60,700 lbs. with 12·7 per cent. extension, and at 400°, 56,750 lbs. with 14·6 per cent., the elastic limit being then as high as 47,000 lbs.

Aich's Metal, which at 60° Fah. has a tensile strength of 57,300 lbs. and 49,600 lbs. at 212° Fah.; it has still 35,500 lbs. at 482° Fah.

Aluminium, which at 60° Fah. has a tensile strength of 26,000 lbs., has at 212° Fah. 21,280 lbs., at 300° Fah. 18,100 lbs., at 390° Fah. 14,100 lbs., and at 480° 10,750 lbs.

Manganese Bronze.—Parson's No. 6 metal, which has a tensile strength of over 76,000 lbs., with an ultimate extension of 25 per cent. at 100° Fah., still has 56,000 lbs. at 450° Fah., with an extension of 27 per cent., and is specially designed for boiler and cylinder mountings and other castings exposed to high temperatures.

CHAPTER XXVI.

OILS AND LUBRICANTS—ENGINE FRICTION.

In these days of large engines and high-speed engines the quality of the lubricant used is of first importance. Notwithstanding this, it is by no means an uncommon thing to find that a shipowner, after paying a considerable price to have a first-class engine, will supply the engineer with an inferior oil for the sake of saving 2d. a gallon. The falseness of this economy is apparent when it is known that there may be a difference of as much as 5 per cent. in the efficiency of an engine by using an inferior oil as against a good one, and that with inferior oil the wear and tear of brasses, &c., may be also serious, whereas, with a proper supply of a good lubricant there should be really no wear or tear. At the same time, it must be admitted that the method of lubrication generally observed on board ship is a primitive and likewise very wasteful one. Surely some system of continuous circulation of the same oil might be followed with the same good results and accuracy as obtain with electrical engines on shore. But it is quite true, however, that to attain this end the use of water on the bearings must first be stopped entirely as a general practice, and only resorted to on emergency; no doubt this practice can be safely followed if the oil is passed through the bearings, pins, &c., as lavishly as is the case with the electrical engine. In the meanwhile it is quite possible and comparatively easy to devise a system whereby the oil from the bearings can be collected, separated from the dirt and water, and so dealt with as to be used over again; it could consequently be supplied much more freely than is the case now with the lubricating gear at present fitted to modern machinery.

Oils are of three kinds—animal, vegetable, and mineral—and have probably been discovered and used by mankind in this order. The two former are called fixed oils because they will not volatilise without decomposition as do the mineral.

Of Animal Oils, the ones most used by engineers are, neat's foot, lard, whale, and sperm.

Sperm Oil is obtained from the head of the sperm whale, and is a very high-class lubricant, in fact, there is none better for guides and bearings, &c., ot engines of all sizes. It is, however, of high price, and there are other mixed oils almost as good for marine purposes to be obtained at a much lower price.

Whale Oil, better known in other days as train oil, is not often used by itself as a lubricant, although, of course, it could be on an emergency if properly refined so as to be free from any substance that would easily solidify and become hard like a gum. It is now generally used for tempering large steel forgings, &c.

Lard Oil has not been used on board ship in this country as a lubricant, but in America, as also nowadays in this country, it is known as a good lubricant, having a viscosity equal to that of olive oil, and found to be the best oil for using on the quick cutting modern machine tools when being forced to their utmost power.

Neat's Foot Oil, obtained from the fat of the feet and hocks of bullocks, was a favourite one with older engineers, and is still used by oil manufacturers to mix with other oils, generally mineral, so as to form a lubricant suitable for general purposes.

Vegetable Oils.—The ones best known to engineers are:—Olive, castor, colza, rape, palm, cotton-seed, linseed.

Linseed Oil, obtained from the seed of flax, is not suitable for a lubricant, but is very useful to the engineer in other ways.

Cotton Oil is expressed from the seed of the cotton plant; when refined it is colourless and tasteless. It may be used instead of lard oil for cutting tools, as also for mixing with other oils. It is rather too volatile for ordinary marine purposes, and, in fact, from the tendency to dry up, it is hardly looked upon as a safe lubricating oil.

Palm Oil, in the form of grease, was formerly sometimes used for the outer bearings of paddle ships, and for the filling of the centre recess of tunnel shafting bearing blocks. It may still be used for these purposes instead of tallow on stations where it is cheap. It is, however, now hardly in the purview of the marine engineer as an every-day lubricant.

Rape and Colza Oils, expressed from the seed of rape or wild turnip, are very much liked, and used generally by locomotive engineers; apparently generally for the reason that, while being good lubricants they also are good lamp oils and not liable to the dangers of fire as is petroleum, for the latter purpose when an accident occurs, a matter of great consideration before the days of train lighting by gas.

Castor Oil, extracted from the castor oil bean, is largely used on board ship in the East Indies where it is cheap as well as plentiful. It is a very good lubricant, and may be used with advantage, especially in hot climates, where it, as other oils, are apt to run thin. It may here be mentioned that the lubricating value of an oil depends very much on the temperature at which it works.

Olive Oil, expressed from the fruit of the olive tree, has been used as a lubricant on board ship very extensively, and is a most satisfactory one, being suitable to bearings and guides of all sizes. At one time it was the only oil used in the British Navy, and when replaced by Rangoon oil by an economy-seeking Board, it was shown to be not so extravagant as the price seemed to indicate. Olive oil solidifies at 36° Fah., and, like castor, has good viscosity at fairly high temperatures.

Mineral Oils.—These oils, quite unknown to the old engineers, are now in very general use both for internal and external lubrication, and seem destined to oust other oils from the marine engine-room. The chief supplies are from Southern Russia (Baku) and from the United States of America, although petroleum is known to exist in almost every part of the world. In this country we have a somewhat similar article in our Scotch shale oils, obtained by the distillation of shale and separated from the paraffin contained therein.

The petroleum, as obtained from the wells, is a thick, treacly substance, Russian having a specific gravity as high as 0·938, and the American 0·886. After refining, the residue known in Russia as *Astatki*, is used for fuel on board ship and on locomotives. The American residuum has a specific gravity of 0·928, and is also used for fuel.

Between that and the highly inflammable light oils, such as petrol, benzaline, &c., are a large number of grades having varying densities as well as viscosities and distilled at different temperatures. Among them are oils suitable for the heavy bearings of marine engines as well as the light ones of the smaller kinds; also oils whose evaporating point is high and therefore suitable for the internal lubrication of marine engines. All these refined oils are pure hydrocarbons—that is to say, they are composed only of hydrogen and carbon. They will not saponify in the ordinary way nor form free fatty acids at all, and therefore may be used safely for internal lubrication without fear of damage to the castings or to condenser and boilers. At the same time, it is, of course, advisable to prevent the passage of any oily matter whatsoever to the boiler, whether it be some of the pure oil carried along without destruction, or the refuse after use in the engine.

The following tables give the leading characteristics of the above oils:—

TABLE LXII.—Specific Gravity of Oils when at 60° F.

Archbutt and Deeley.

Sperm oil,	0·880
Neat's foot oil,	0·915
Lard oil,	0·916
Whale oil,	0·922
Seal oil,	0·925
Porpoise oil,	0·926
Olive oil,	0·915
Rape-seed or Colza oil,	0·915
Cotton-seed oil,	0·923
Castor oil,	0·965
Palm oil,	0·923
Rosin oil,	0·980
Scotch lubricating oils,	0·875 to 0·895
American dark machinery oils,	0·877 ,, 0·887
,, light ,,	0·897 ,, 0·920
,, cylinder ,,	0·885 ,, 0·905
Russian pale for light machinery oils,	0·895 ,, 0·903
,, heavy ,,	0·903 ,, 0·910

TABLE LXIII.—Density of Oils at Various Temperatures.

Archbutt and Deeley.

Name of Oil.	60° F.	100° F.	150° F.	212° F.
Sperm oil,	0·8783	0·8637	0·8455	0·8229
Olive oil,	0·9159	0·9011	0·8826	0·8596
Rape or Colza oil,	0·9151	0·9005	0·8822	0·8595
Castor oil,	...	0·9473	0·9284	0·9050
Scotch mineral oil, "865,"	0·8683	0·8533	0·8347	...
,, "890,"	0·8905	0·8761	0·8581	...
Russian mineral light machinery oil,	0·8978	0·8837	0·8661	0·8442
,, heavy ,,	0·9096	0·8957	0·8784	0·8568
American mineral spindle oil,	0·8677	0·8535	0·8358	0·8138
,, red engine oil,	0·9162	0·9020	0·S843	0·8624
,, Bayonne engine oil,	0·9113	0·8973	0·8797	0·8579
,, dark medium machinery oil,	0·8839	0·8695	0·8514	0·8291
"Valvoline,"	...	0·8757	0·8587	0·8377
Dark filtered cylinder oil, "900,"	...	0·8883	...	0·8490

The Flash Point of American cylinder oil (sp.gr., 0·902) is 585° F.
" " " (" 0·893) is 424° F.
" " machinery oil (" 0·897) is 402° F.
" Russian " (" 0·909) is 380° F.

The Boiling Point of mineral oils is unusually high, and in the case of refined distilled oils seldom less than 600° F., that of linseed oil being 602° F. The American oils are seldom fluid at a temperature under 25° F., while the Russian generally remain so as low as 0° F.

Rape oil solidifies at freezing point (32° F.), lard oil at 40° F., sperm oil at 39° F., and olive oil at 36° F.

TABLE LXIV.—VISCOSITY OF OILS AT VARIOUS TEMPERATURES (WATER AT 68° F. BEING 0·01028)—*Archbutt and Deeley.*

Name of Oil.	60° F.	100° F.	150° F.	212° F.
Sperm oil,	0·420	0·185	0·085	0·046
Olive "	1·008	0·377	0·154	0·070
Rape or Colza oil,	1·118	0·422	0·177	0·080
Castor oil,	...	2·729	0·605	0·169
Scotch mineral oil "865,"	0·146	0·066	0·036	...
" " "890,"	0·509	0·183	0·069	...
Russian light machinery oil,	1·156	0·307	0·099	0·043
" heavy "	3·592	0·762	0·196	0·066
American spindle oil,	0·727	0·236	0·086	0·039
" pale " "903/7,"	1·138	0·342	0·115	0·049
" " " "907/12,"	1·479	0·419	...	...
" red engine oil "910/20,"	1·915	0·496	0·150	0·058
" "Bayonne" engine oil "912/15,"	2·172	0·572	0·173	0·063
" dark medium machinery oil,	3·046	0·705	0·210	0·076
"Valvoline,"	...	2·406	0·605	0·187
American filtered cylinder oil "888,"	...	3·702	...	0·238
" dark " " "900,"	...	6·264	...	0·314

TABLE LXV.—BOILING, SETTING, AND FLASH POINTS.

Name or Brand of Oil.	Sets at	Flashes at	Boils at
	Degrees F.	Degrees F.	Degrees F.
American cylinder, sp. gr. ·902,	25	585	650
" " " ·893,	25	424	600
Valvoline "	...	...	700
American machine, sp. gr. ·897,	25	402	600
Russian " " ·909,	0 to 10	380	...
Scotch shale oil,	32	...	...
Cotton-seed oil,	34 to 50	...	580
Castor oil,	14	...	556
Lard oil,	25 to 42	...	..
Linseed oil,	5 " 17	...	574
Neat's foot oil,	32 " 50	...	...
Olive oil,	21 " 39	...	...
Rape or Colza oil,	10 " 28	...	625
Sperm oil,	32	450	600
Palm oil,	81 to 106	...	...
Whale oil,	32	440	620
Vaseline,	104 to 120	640	...
Tallow (Russian),	107 " 119	540	660

TABLE LXVI.—Price per Gallon of Various Oils.

J. L. Seaton & Co., Hull.

Name of Oil.	Price per Gallon.	Name of Oil.	Price per Gallon.
Castor oil, ...	2/3 to 2/6	Palm oil, ...	2/9 to 3/-
Colza oil, ...	2/1	Sperm oil (common), . .	2/3 „ 2/6
Rape oil, ...	2/-	Whale oil, ...	2/-
Cotton-seed oil, . . .	2/-	Neat's foot oil, . . .	3/- to 3/3
Linseed oil, . . .	2/- to 2/3	Lard oil, ...	4/- „ 4/3
Olive oil, ...	3/3 „ 3/9	American lubricating (good),	1/3 „ 1/6

Compound Oils.—The oils most generally used on shipboard for external lubrication are compounded of a mineral oil and one or more of the fixed oils. Mineral oil of itself, although, on the whole, a good lubricant, is not suitable for bearings or guides which have, or may have, a greater pressure per square inch than 70 lbs. Moreover, increasing the viscosity of a mineral oil by other denser oils does not improve it for the purpose, and, in fact, is often the means of causing greater friction. If, however, a fixed or fatty oil be compounded with the mineral oil in a proper way so that they do not separate out when standing still, the compound will form a suitable lubricant for higher pressures than the above. The higher the pressure per square inch, the greater viscosity must be given to the lubricant by adding fatty oils of higher viscosity. It will also be observed by reference to Table lxiv. that mineral oils whose viscosity is very high at 60° Fah. are the very ones whose viscosity rapidly decreases with an increase of temperature, and although the fatty oils also decrease in viscosity as the temperature rises, their rate of fall is not so great. In other cases, however, pure mineral seems to be lacking in what Messrs. Archbutt and Deeley term "oiliness."

Mineral oils are sometimes thickened in viscosity by dissolving certain soaps in them, the favourite for this purpose being aluminium soap, which is made by saponifying whale, cotton, or even lard, oils and caustic soda, and pouring into it when hot a solution of common alum. This is, of course, an adulteration, but not altogether a bad one, inasmuch as the oil then sticks to the journal much better than it did in the pure state and gives a better lubrication to the bearing. The soap, however, is liable to start out and obstruct the oilways. When mineral oil has been treated in this way it can generally be detected by the tendency to threadiness when the cork is drawn from the bottle of the mixture.

The oils most generally used for compounding with mineral oils are:—Rape, olive, lard, neat's foot, whale, and sperm. Castor oil will not dissolve in and compound with a mineral oil unless it is first treated with an equal volume of tallow oil. It is, however, not often used for the purpose. Rosin oil is sometimes used to increase the specific gravity of a compound oil as well as to improve its viscosity. It does not, however, improve its lubricating powers, and as it increases the tendency to gumminess it should be avoided. Marine engineers usually prefer the compound oil to a pure mineral one on account of the tendency to lather—that is, to form grease by mixing with the water applied to the bearing. This formation is hastened, of course, by the use of a little soda in the water, and when formed it generally prevents the oil running freely out of the bearing or guide. Besides which, should the bearing or guide become warm, this grease melts and adds to the quantity of lubricant so as often to check any further heating.

Soaps.—Pure mineral oils are unaffected by alkalies and have little or no effect on metals. They therefore do not saponify, and for this reason are used for internal lubrication. Formerly, when tallow and fatty oils were used indiscriminately for internal and external lubrication, the condenser very soon got covered with a soapy deposit more or less hard. All vegetable and animal oils contain fatty matters, which combine with alkalies and form what are called soaps. Some oils will also combine with certain salts of metals and form a soapy mixture. For example, white lead and linseed oil. Oils combined with caustic soda make hard soaps, and, when combined with caustic potash, form what is known as soft soap. Solutions of these soaps will act as lubricants, but are, of course, not so efficient as the oils. They are sometimes temporarily used to cleanse oil ways of the gummy greasy matter which may have been deposited from the oil.

Adulteration of Oils.—Compound oils are generally valued by their density and viscosity. It is therefore the aim of the dishonest merchant to add some cheap ingredient which shall effect this object. That most generally used for the purpose is resin oil, being cheap and easily mixed with other oils. It is, however, itself inferior as a lubricant, and has the fatal vice of drying hard, like varnish. Its presence in oils is easily detected by the pungent taste, which is unmistakable,* and its failure to saponify when an alkali is added.

Solid Lubricants.—There are certain substances which act as lubricants, although not at all on the same principle as does a liquid. The chief of these are powdered soapstone, flour of sulphur, and plumbago.

Soapstone and French chalk, finely powdered to a flour, has a greasy feel and acts as a very efficient lubricant to wood fittings, and also, to some extent, to metallic ones. It is, however, of little practical value as a modern engine lubricant.

Flour of Sulphur was frequently used by the older engineers, whose acquaintance with hot bearings was much more intimate and frequent than is that of their successors to-day; they always carried it among their stores, and, when a bearing was troublesome, poured a small quantity down the oil holes. Its efficiency probably varied with the heat of the bearing. Its function seems to have been to combine with the brass dust and brass ridges, and thereby to form a sulphide or sulphides of the metals which was much softer than the metals themselves, and, in a plastic state, permitted of the formation of a web which lined the bearing and formed a new surface for the journals to run on. If this theory is a correct one, the material, of course, cannot be considered as a true lubricant.

Graphite or Plumbago.—This material, when pure and so highly subdivided that it forms an impalpable powder, has an extremely greasy feel to the fingers, and is an exceedingly good lubricant for both metals and woodwork. It may be used together with oil, grease, or vaseline for large and heavy bearings revolving at moderate velocity. It is also used for the internal lubrication of some engines, and doubtless may, in the near future, play an important part in that direction when superheated steam is used on shipboard. At present the means of applying it to the internal parts are crude and unsatisfactory. Ingenuity, however, will soon find a method to overcome the present difficulties, and, whether this material is carried over in a dry state or mixed with vaseline or some other semi-liquid carrier, it will be properly distributed and do good service. A small quantity of graphite may be used with advantage on bearings which are lubricated with oil, as it tends to keep the surfaces of both journal and bearing smooth and polished. When a brass has become rough or scored, this material will fill the recesses and present a new and good surface to the journal.

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* For further particulars of oils, see *Lubrication and Lubricants*, by Archbutt and Deeley, published by Griffin & Co.

Tallow and certain other fats, both vegetable and animal, are, at ordinary temperatures in temperate zones, in the solid state, but a slight rise of temperature converts them into a liquid, and therefore it is idle to deal with them as if they were solid lubricants. Moreover, none of them are of great importance to the marine engineer of to-day.

Grease.—If oils whose specific gravity is nearly that of water be mixed with pure warm water and agitated, an *emulsion* is formed, and the oil will separate from the water very slowly indeed, even when the mixture is allowed to stand. If the water is made slightly alkaline and carefully stirred into some animal and vegetable oils, especially if at a moderately high temperature, they combine and form grease—the value of which as a lubricant depends in great measure on the quality and quantity of the oil used in making it. Tallow, palm oil, and other such fatty substances will take up very considerable quantities of water.

TABLE LXVII.—Coefficients of Friction.

Nature of the Surfaces.		Quite Dry.	Greasy Dry.	Well Lubricated with			Constant Flow of Oil.
				Oil	Tallow or Soap.	Water.	
Wood on wood (dry),		0·25 to 0·5	...	...	0·2 to 0·04	...	...
Metal on hard wood,		...	0·15	0·07 to 0·08	...	...	0·05
Metals on oak,	(Rk)	0·5 to 0·6	0·24 to 0·26	...	0·2	...	...
,, metals only,	,,	...	0·15 to 0·2	...	0·14	0·3	...
,, ,, smooth,	,,	...	0·07 to 0·08	0·05	...	...	0·03
Bronze on lignum vitæ,	,,	...	...	...	...	0·05	...
Iron ,, ,,		...	...	...	...	0·05	...
Bronze on bronze sliding,	(S)	...	...	...	...	0·38	...
,, ,, ,,	(Rn)	...	0·175	...	...	...	...
Steel ,, ,,	,,	...	0·139	...	...	...	...
Wrought iron on bronze sliding,	,,	...	0·135	...	...	...	...
Cast ,, ,, ,,	,,	...	0·141	...	...	...	...
Steel on cast iron	,,	...	0·151	...	...	...	...

TABLE LXVII.—COEFFICIENTS OF FRICTION—*Continued.*

Nature of the Surfaces.	Quite Dry.	Greasy Dry.	Well Lubricated with			Constant Flow of Oil.
			Oil.	Tallow or Soap.	Water.	
Steel on wrought iron, (Rn)	...	0·189	...	...	...	...
Wrought iron on cast iron, ,,	...	0·170	...	...	...	...
,, ,, tin, ,,	...	0·181	...	...	...	...
,, ,, magnolia, (RHS)	...	0·060	...	...	...	...
,, ,, white metal,	...	...	0·025	...	...	0·01 to 0·015
Loco axles, white metal, (W)	...	...	0·017	...	...	...
Shafts on bronze, (Rn)	...	...	...	0·028	...	...
Leather on metals, (Rk)	0·56	0·23	0·15	...	0·36	...
Wrought-iron shaft in cast-iron bearing, speed 10 feet per minute,	...	...	0·079	...	...	...
Wrought-iron shaft in cast-iron bearing, speed 60 feet per minute,	...	...	0·052	...	...	...
Wrought-iron shaft in cast-iron bearing, speed 100 feet per minute,	...	...	0·050	...	...	...
Load 50 lbs. per square inch, speed 50 feet, (G)	...	...	...	...	...	·0032
,, 50 ,, ,, ,, 110 ,, ,,	...	...	...	...	...	·0064
,, 50 ,, ,, ,, 190 ,, ,,	...	...	...	..	...	·0106
,, 150 ,, ,, ,, 50 ,, ,,	...	...	...	...	...	·0017
,, 150 ,, ,, ,, 110 ,, ,,	...	...	...	...	...	·0024
,, 150 ,, ,, ,, 190 ,, ,,	...	...	...	...	...	·0047
Speed of 10 feet per min., load 50 lbs., ,,	...	...	...	..	...	·0009
,, 10 ,, ,, 75 ,, ,,	...	...	...	...	...	·0007
,, 10 ,, ,, 150 ,, ,,	...	...	...	...	...	·0250
,, 15 ,, ,, 50 ,, ,,	...	...	...	...	...	·0012
,, 15 ,, ,, 150 ,, ,,	...	...	...	...	...	·0051
,, 7-8 feet per minute, moderate load, ,,	...	0·094	0·072	...	..	·064
,, ,, ,, heavy ,, ,,	...	0·051	0·046	...	...	·047
Railway axles and bearings (static),	...	...	0·100	0·067	...	...
Cast-iron steel tyres, (Gl)	0·36 to ·048	5 to 60 miles per hour.				
Steel tyres on steel rails, ,,	0·11 to 0·04	10 to 50 ,, ,,				

Note.—Rk stands for Rankine; Rn, for Rennie; S, for Summers; W, for Wood; G, for Goodman; RHS, for Prof. Smith; Gl, for Gallon and Westinghouse.

Rule.—Let S be the velocity of sliding in feet per minute, P the maximum steady working pressure in lbs. per square inch on a slide lubricated in the common way (that is, without force).

Then

$$P = 50{,}000 \div S + 100$$

if intermittent as in piston-rod guide, maximum value is P, then

$$P = 65{,}000 \div S + 100.$$

For journals with steady load

$$P = 65{,}000 \div d \times R + 30,$$

d being diameter in feet and R the revolutions per minute.

For intermittent circular as in crank-pins

$$P = 100{,}000 \div d \times R + 30.$$

For intermittent oscillating as in cross-head brasses

$$P = 250{,}000 \div R + 130.$$

Engine Friction.—The following figures and diagrams show the results of some carefully made trials with two triple-compound engines, by Messrs. Belliss & Morcom, Birmingham. They are of their special enclosed type, having three cylinders and three cranks supplied with steam from Babcock & Wilcox boilers at 150 lbs. pressure, and exhausting into a surface condenser whose pumps are operated by independent engines. The whole of the pins, bearings, guides, and working parts are lubricated with their special American mineral oil forced through them by a pump constantly at work connected to the engine. It will be seen that the friction per revolution, both with and without load, varies, and has two minimum and two maximum values; that at full speed the frictional H.P. per revolution loaded is really only a little less than that when running free, whereas at 200 revolutions, or half speed, there is the greatest difference between them, that at full load being half that running free; also that at dead slow speed, say 25 revolutions, the friction per revolution, both light and loaded, is nearly four times that of the loaded engine at 200 revolutions.

ENGINE (No. 1401) $\frac{11'' - 17'' - 24''}{12''}$ 150 LBS. AT ENGINE STOP-VALVE.

Table of Powers, &c., Under Load Conditions.

	Revolutions.			
	166	300	350	400
I.H.P.,	248·2	384·15	408·1	429·4
B.H.P.,	240·4	360·5	380·5	400
Difference or friction, H.P.,	7·8	23·65	27·6	29·4
I.H.P. per revolution,	1·496	1·280	1·168	1·073
B.H.P. " "	1·45	1·201	1·086	1·00
F.H.P. " "	·047	·079	·079	·073
$\frac{\text{B.H.P.}}{\text{I.H.P.}}$ efficiency,	96·8 %	93·9 %	93·25 %	93·15 %

Friction Powers Under No-Load Conditions.

	Revolutions.					
	56	102	113	213	313	365
Friction H.P.,	4·9	10·7	7·4	24·6	36·1	30·9
" " per revolution,	·0875	·1049	·0655	·1153	·1153	·0847

The above figures are embodied in the accompanying diagram.

The no-load cards were taken non-condensing, but the load cards were taken condensing.

This is probably the reason why the friction power shown at no-load is greater than that at full-load. The efficiency under load condensing is invariably slightly better than when non-condensing.

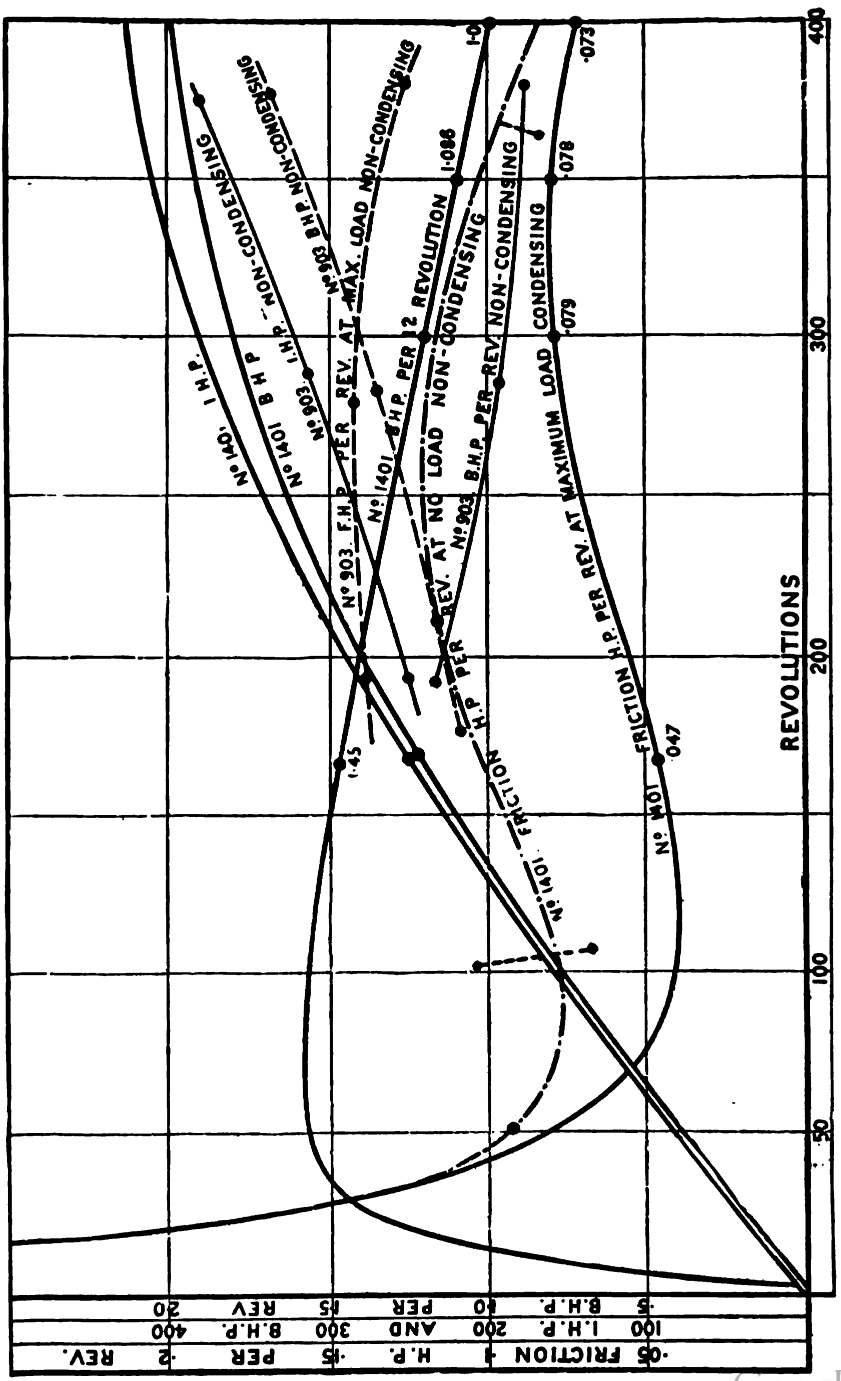


Fig. 231.—Friction Trials with a Triple-Compound Engine by Belliss & Morcom.

(1401) Cylinders 11 – 17 – 24 inches diameter × 12 inches stroke, steam 150 lbs., cut-off 0·8.

(903) ,, ,, ,, 10½ ,, ,, ,, ,,

No. 903, a similar engine, but 10½ inches stroke instead of 12 inches, ran non-condensing with the following results.

	Revolutions.		
	192	285	380
I.H.P., ...	249·8	312·6	387·9
B.H.P., ...	223·2	273·5	340
F H.P., ...	26·6	39·1	47·9
I.H.P. per revolution, .	1·3	1·096	1·02
B.H.P. ,, ,, .	1·162	·96	·895
F.H.P. ,, ,, .	·1385	·142	·126
$\frac{\text{B.H.P.}}{\text{I.H.P.}}$ efficiency, . .	89·3 °/。	87·5 °/。	88 °/。

The result *condensing* at two different pressures is—

	Revolutions.	
	380	380
I.H.P., ...	392·6	496·9
B.H.P., ...	357·5	462·4
F.H.P., ...	35·1	34·5
I.H.P. per revolution, . .	1·033	1·305
B.H.P., ...	·942	1·215
F.H.P., ...	·0925	·0909
$\frac{\text{B.H.P.}}{\text{I.H.P.}}$ efficiency, ...	91·4 °/。	93 °/。
Steam pressure, ...	Gauge — Engine 155 - 128	Gauge — Engine 170 - 161

APPENDIX A.

BREMME'S VALVE-GEAR.

I AM indebted to Mr. Gustav A. C. Bremme, of Liverpool, for the following investigations:—

The accompanying diagrams (figs. *a* and *b*) represent the Gear as applied to Inverted Marine engines. It is here assumed that the engines are worked from the starboard side, so that fig. *a* shows the ahead position and fig. *b* the astern position of the Gear.

If the engines are worked from the port side, fig. *b* shows the ahead and fig. *a* the astern position of the Gear, as is easily understood.

The eccentric coincides with the crank, and its radius $= r$.

K H $= l$ is the stiff eccentric-rod, jointed at H to the radius-rod, H G $= \rho$, which swings on G, the gudgeon or joint attached to F G $= \rho$, the radius-arm, which is movable on the fixed centre, F.

H E $= m$ is a prolongation of the eccentric-rod, K H, with which it may, if desirable, form a slight angle, and which is termed the "Lead-arm," as the amount of lead is proportionate to its length.

The valve-rod is jointed at E, and the distances, E N $= \sigma$, practically represent the oscillations of the valve.

The angle, δ, at which the radius-arm deviates on either side from the vertical, through the fixed centre, F, and which is termed the "deviation angle," should never exceed 25°.

The crank-shaft moves in the same direction in which the radius-arm, F G, deviates from the vertical through F.

The distance, O F $= a$, between centre of crank-shaft and fixed centre, F, is $= (l^2 - r^2)^{\frac{1}{2}}$, so that, when the crank-shaft is at either dead-point, the centre, H, covers the fixed centre, F. Hence the lead $= \frac{m}{l} \cdot r$ remains constant for all grades of cut-off.

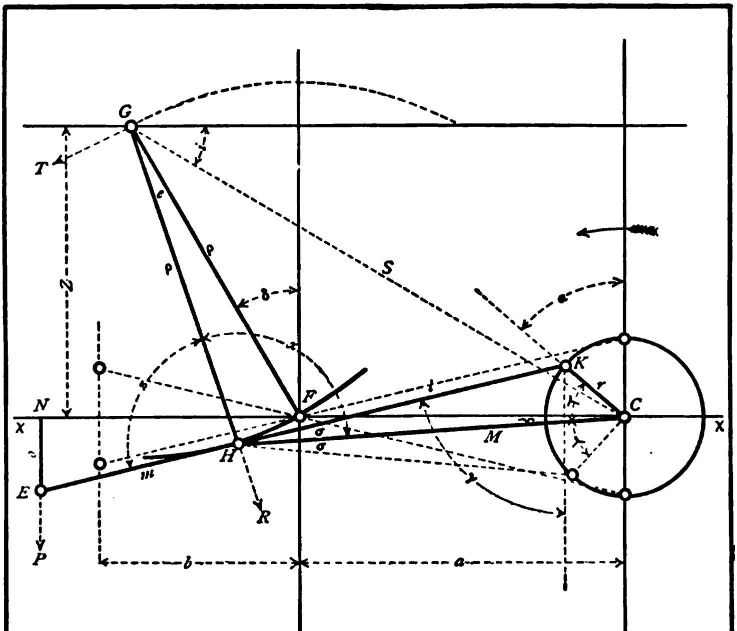
As the centre, H, travels through an arc described by the radius-rod, H G, the oscillations, σ, are greater above than those below the centre line, $\chi\chi$. This difference between the upper and lower oscillations, which is affected by the length of radius-rod and arm, has the advantage that the valve-openings are less, and the cut-offs and cushionings are earlier, for the down-stroke than for the up-stroke, so that the momentum of the moving weights is always suitably balanced.

To determine the best proportions of gear, it is necessary to construct Tables, giving the values of N E $= o$ for a sufficient number of crank-angles, by the following method:—

$$\text{General formula for oscillation, } o = (l + m)\cos\gamma - r\cos\alpha.$$

Count all the crank-angles, α, from 0° to 360°, from the upper dead-point, and reckon the values of σ positive (+) below the centre line, $\chi\chi$, and negative (−) above.

Fig. a.


G
T
z
e
ρ
ρ
δ
S
α
K
r
C
N
X
X
F
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σ
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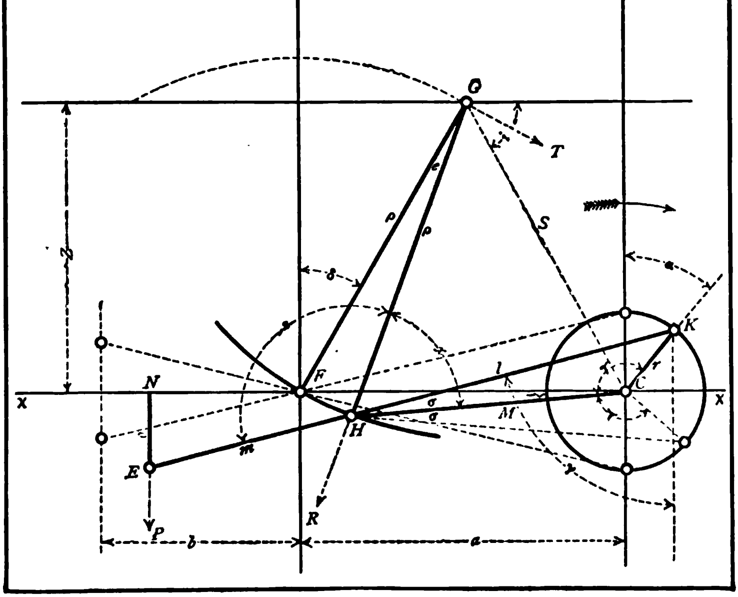
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Fig. b.

For the given values of a, ρ, and angle, δ, find the values z, s, and angle, i—

$$s^2 = a^2 + \rho^2 \pm 2\,a\,\rho \sin \delta\,; \quad z = \rho \cos \delta\,; \quad \sin i = \frac{z}{s}.$$

The values of C H = M range from a to $(l + r)$ and from a to $(l - r)$.

Find angles, τ and σ, for the given values of M, l, and r—

$$\cos \tau = \frac{M^2 + r^2 - l^2}{2\,M\,r}\,; \quad \sin \sigma = \frac{r}{l} \sin \tau.$$

Each value of M corresponds to two crank-angles.

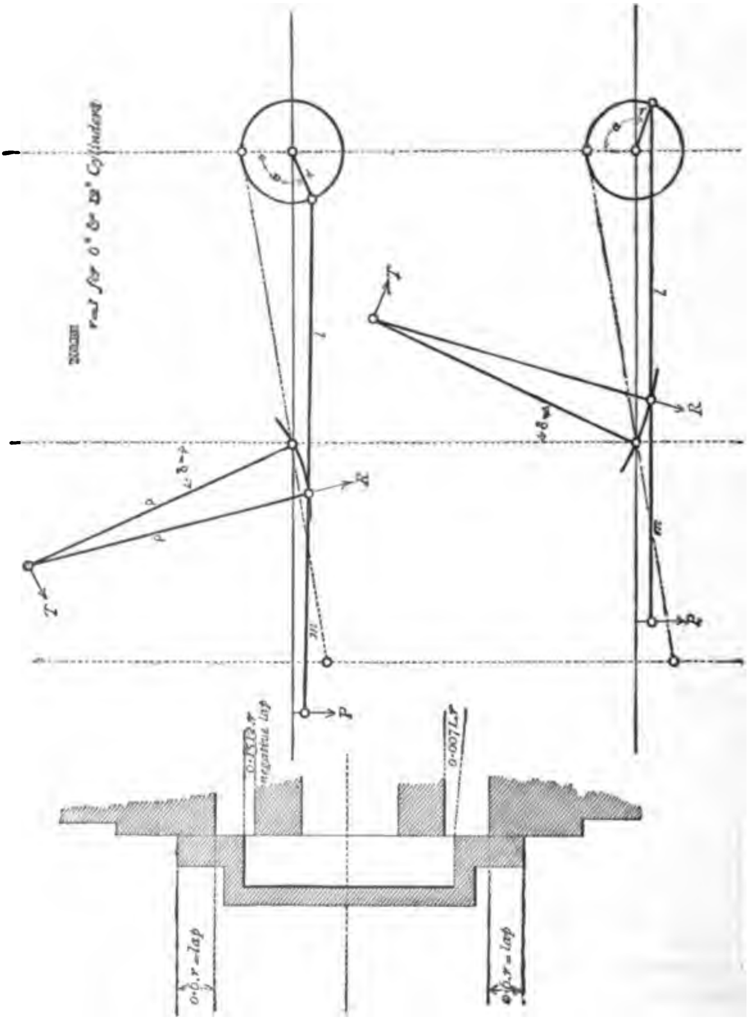

Fig. 6

TABLE LXVIII.

NOTE.—The Piston-rod = 4½ Cranks. (See fig. *a*.)	δ = 25°.				δ = 22½°.				δ = 20°.			
	Down.		Up.		Down.		Up.		Down.		Up.	
	Δ α	Pist. pos.	Δ α	P.	Δ α		Δ		Δ α		Δ α	
Lead opening, .	0° 0′	0·15 *r*	0° 0′	0·15 *r*	0° 0′	0·15 *r*	0° 0′	0·15 *r*	0° 0′	0·15 *r*	0° 0′	0·15 *r*
Maximum opening,	35° 58′	0·4166 *r*	56° 29′	0·6412 *r*	35° 44′	0·3722 *r*	45·57	0·5507	35·29	0·3251 *r*	46·25	0·4726 *r*
Cut-off, . .	95·2	0·6 *St.*	110° 40′	0·626 *St.*	88·31	0·546 *St.*	105·18	0·589 *St.*	81·31	0·448 *St.*	100·52	0·548 *St.*
Cushioning, . .	134·9	0·88 *St.*	145° 27′	0·895 *St.*	132·57	0·87 *St.*	142·54	0·875 *St*	125·17	0·825 *St.*	139° 16′	0·851 *St.*
Release, . .	126·22	0·83 *St.*	138·4	0·85 *St.*	120·26	0·8 *St.*	134·6	0·82 *St.*	115·27	0·764 *St.*	131° 0′	0·8 *St.*

NOTE.—The Piston-rod = 4½ Cranks. (See fig. *b*.)	δ = 25°.				δ = 22½°.				δ = 20°.			
	Down.		Up.		Down.		Up.		Down.		Up.	
	Δ α	Pist. pos.	Δ α		Δ α		Δ α		Δ α		Δ α	
Lead opening, .	0° 0′	0·15 *r*	0° 0′	0·15 *r*	0° 0′	0·15 *r*	0° 0′	0·15 *r*	0° 0′	0·15 *r*	0° 0′	0·15 *r*
Maximum opening,	30·28	0·4184 *r*	59° 55′	0·6438 *r*	30° 45′	0·3699 *r*	59·30	0·5638 *r*	31·1	0·3242 *r*	59·0	0·4823 *r*
Cut-off, . .	90·1	0·56 *St.*	120·14	0·71 *St.*	84·0	0·51 *St.*	114·43	0·656 *St.*	77·15	0·432 *St.*	109·21	0·61 *St.*
Cushioning, . .	129° 48′	0·85 *St.*	150·49	0·925 *St.*	125·58	0·83 *St.*	148·35	0·908 *St.*	121·58	0·802 *St.*	145·47	0·895 *St.*
Release, . .	120·54	0·81 *St.*	144·10	0·885 *St.*	115·32	0·764 *St.*	141·45	0·87 *St.*	110·56	0·73 *St.*	138·26	0·85 *St.*

NOTE.—The crank-angles, α, are counted here from both respective dead-points.
Maximum strains (at α = 56° 29′ in the up-stroke, fig. *a*, and δ = 25°); R = 2, 16 P; T = 0·3377 P.

Find angle, F O H = ξ, for the given values of M, S, and ρ—

$$\cos(i \pm \xi) = \frac{M^2 + S^2 - \rho^2}{2MS}.$$

Then the values of the angles, α and γ, for the four quadrants of the circle are as follows:—

For the ahead position, fig. *a*—

$$\alpha = 90 + \xi - \tau; \quad 90 + \xi + \tau; \quad 90 - \xi + \tau; \quad 450 - \xi - \tau.$$
$$\gamma = 90 - \sigma - \xi; \quad 90 + \sigma - \xi; \quad 90 + \sigma + \xi; \quad 90 - \sigma + \xi.$$

For the astern position, fig. *b*—

$$\alpha = \tau - 90 - \xi; \; 270 - \xi - \tau; \; 270 + \xi - \tau; \quad 270 + \xi + \tau.$$
$$\gamma = 90 - \sigma - \xi; \quad 90 + \sigma - \xi; \quad 90 + \sigma + \xi; \quad 90 - \sigma + \xi.$$

Table lxviii. gives the results for the following proportions of the gear and valve:—

Radius-rod and radius-arm, ...	$\rho = 6\,r$.
Eccentric-rod, K H, ...	$l = 6\,r$.
Lead-arm, H E, ...	$m = 4{\cdot}5\,r$.
Lap on steam-edges (both ends equal) . . as shown by fig. *c*.	$= 0{\cdot}6\,r$,

NOTE.—As a general rule, the lap on the steam-edges may be $0{\cdot}75 \frac{m}{l} r$.

The chief feature of Morton's Reversing "Valve-Gear" is the method employed for correcting the irregular angularity of the connecting-rod, and the maintenance of the correction whether the engines are set to work in full forward or backward gear or any intermediate position. Equal port-openings for both ends of the cylinder are thus given, if necessary: the lead is constant for all grades of expansion, and the motion of the slide-valve is such that "wire-drawing" and "cushioning" are reduced to a minimum, the motion of the valve being in unison with the stroke of the main piston instead of being in unison with the revolution of the crank, as is the case with other valve-gears of this class.

The movable point, A, receives its motion from the crosshead, as shown by the diagram (fig. *d*), and radiates with the short lever or crank, P, centred in a projection, P, on the connecting-rod, receiving its motion from a point in the valve-lever, F. The proportions are such that the versed sine of the crank, P, will be equal to the versed sine of the lever, F, as is customary with this system when used as a parallel motion. The driving end of the lever, F, is connected with the crosshead by means of the link, B, and its fulcrum end, G, is suspended by links, C, vibrating from fixed centres, K, at their upper ends. These links are always parallel with the connecting-rod when the engine is "on the centre." The radiating movable point, A, corrects the angularity, both of the connecting-rod and of the valve lever, F, by one motion. The movement of the links, C, in vibrating from right to left or left to right from the centre line, in whatever direction the crank may revolve, is equal for equal increments of the travel of the piston. These conditions are maintained whether the engines are in forward or in backward gear or linked up to any intermediate position. The reversing portion of the gear consists of a quadrant, I, properly secured so as to form part of the slide valve-rod and to move with it. The bracket forming the guide for the valve, spindle, and quadrant has arms cast on, which carry the fixed centres, K, of the vibrating links, C.

The arc of the quadrant, I, being the radius of the adjustable link, H. from the centre, N, is fitted with a die block, J, capable of being moved to the right or left from the centre of the quadrant for forward or backward motion of the engines; consequently, when the engine is "on the centre," the upper end of the adjustable link, H, and its block, J, may be moved through the whole range of the quadrant, I, from right to left, or *vice versâ*, without moving the slide valve.

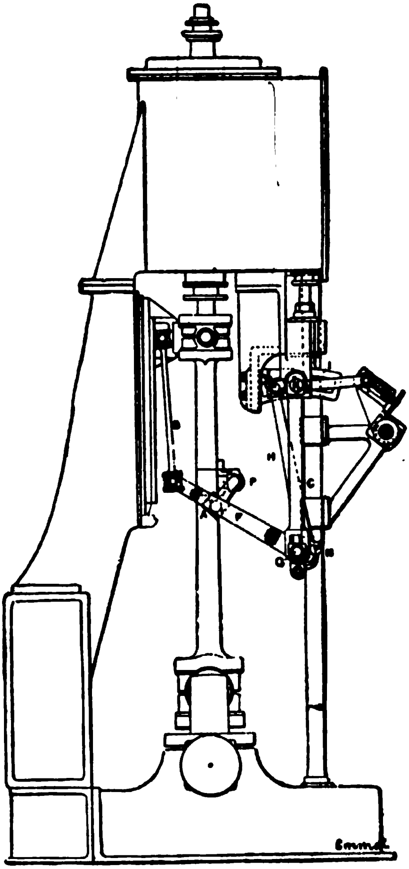

Fig. *d.*—Morton's Valve-Gear.

This arrangement reduces to a minimum the wear of the die block in the quadrant, for, when it is once set in position for working, there is little or no motion.

APPENDIX B.

Cotterell's Method of Constructing Inertia Curves.

An acceleration curve may be constructed by graphical differentiation of the velocity curve, a method which it may be worth while to illustrate in detail.

Divide the crank pin circle into a number of equal parts, and, supposing the connecting-rods drawn, let them cut the vertical through 0 in the points 1′, 2′, 3′, in fig. *e*. Also find and mark off the corresponding positions of the piston, 1″, 2″, 3″, &c. Now, since the lengths 0 1′, 0 2′, 0 3′, &c., represent the velocities of the piston and reciprocating parts when in positions 1″, 2″, 3″, &c., the difference between any two consecutive lengths, for example, 1′, 2′, will represent the change of velocity that has taken place in the corresponding movement of the piston, 1″, 2″. If we suppose the crank pin to revolve uniformly and divide the circle into equal parts, equal times will be occupied in the motions from point to point, and, therefore, equal times in the motions between consecutive positions, 1″, 2″, 3″, 4″, &c., of the piston. Accordingly, the differences, 0 1′, 1′ 2′, 2′ 3′, &c., will represent the force required to change the velocity of the reciprocating parts; and if we set them up as ordinates between the corresponding positions of the piston we shall obtain the curve expressing the effect of inertia. The ordinate should be erected from the position of the piston when the crank-pin is at the middle of the intervals, 1, 2, 3, &c.

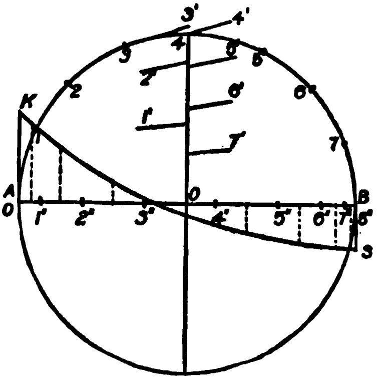

Fig. *e*.

It will be seen that the greater the number of parts into which we divide the crank-pin circle the less will be the ordinates representing the effect of inertia, though in all the curves the same character will be preserved. Accordingly, it is possible to determine the number of parts into which the crank circle should be divided, or to determine the angle between consecutive radii, 0 1, 0 2, &c., so that the ordinates of the inertia curve may be of a length proper to represent the pressure per square inch of piston area required for inertia, on the same scale that the indicator diagram is drawn. The ordinates of the resulting inertia curve may then be directly employed to correct the indicator diagram.

Let N be the number of revolutions per minute; Q, Q^1 (fig. *f*) consecutive points on the crank-pin circle; and let $QOQ^1 = n^0$ be the required angle. Further, suppose that the crank-pin circle is drawn on a scale of x inches to the foot. Then the change of velocity of piston, Δv, in feet per second is evidently—

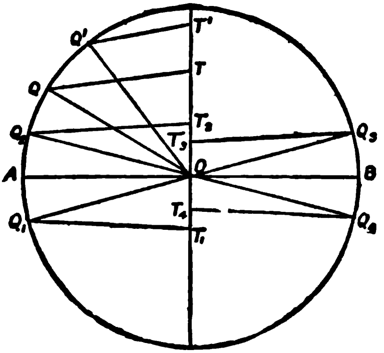

Fig. *f*.

$$\therefore \quad \Delta v = \frac{2\pi N}{60}\,\frac{T\,T^1}{x},$$ where $T\,T^1$ is to be measured in inches.

Now, this change of velocity takes place in the time, Δt, occupied by the movement through n^0—that is, in $n/6$ N seconds.

$$\therefore \quad \text{Force due to inertia} = \frac{W}{g}\,\frac{\Delta v}{\Delta t} = \frac{W}{g}\,\frac{2}{10}\,N^2\,\frac{T\,T^1 \text{ in inches}}{x\,n^0}.$$

And if the indicator diagram is drawn on the scale of y lbs. to the inch, it will be found that—

$$n^0 = \cdot 0195\,p_0 \cdot \frac{N^2}{x\,y}.$$

The curve will cross the base line at the point L, where the piston has its maximum velocity—that is, approximately, when the crank is at right angles to the connecting-rod—hence

$$O\,L = \sqrt{(\text{connecting-rod})^2 + (\text{crank})^2} - \text{connecting-rod}.$$

APPENDIX C.—SPARE GEAR.

BOARD OF TRADE RULES.

Spare Gear and Stores to be carried.—Foreign-going steamers coming in for survey must be provided with spare gear. In the case of steamers coming in for survey under the Passengers Acts, and other steamers performing ocean voyages, no question as to gear need be raised if the following spare gear and stores are supplied. The heavier portions of this gear must have been fitted and tried in its place, and must be kept on board where access can at all times be had to it:—

1 pair of connecting-rod brasses.
1 air-pump, bucket, and rod, with guide.
1 circulating pump, bucket, and rod.
1 air-pump head valve, seat, and guard.
1 set of India-rubber valves for air-pumps.
1 circulating pump head valve, seat, and guard.
1 set of India-rubber valves for circulating pumps.
2 main bearing bolts and nuts.
2 connecting-rod bolts and nuts.
2 piston-rod bolts and nuts.
8 screw shaft coupling bolts and nuts.
1 set of piston springs, suitable for the pistons.
3 sets, if of India-rubber, or 1 set, if of metal, of feed-pump valves and seats.
3 sets, if of India-rubber, or 1 set, if of metal, of bilge-pump valves and seats.
1 hydrometer.
Boiler tubes, 3 for each boiler.
100 iron assorted bolts, nuts, and washers, screwed, but need not be turned.
12 brass bolts and nuts, assorted, turned, and fitted.
50 iron ,, ,, ,, ,,
50 condenser tubes.
100 sets of packing for condenser tube ends, or an equivalent.
At least 1 spare spring of each size for escape valves.
1 set of water-gauge glasses.
$\frac{1}{10}$ the total number of fire-bars necessary.
3 plates of iron, assorted.
6 bars of iron, assorted.
1 complete set of stocks, dies, and taps, suitable for the engines.
1 smith's anvil.
1 fitter's vice.
Ratchet-braces, and suitable drills.
1 copper or metal hammer.
Suitable blocks and tackling for lifting weights.
1 dozen files, assorted, and handles for the same.
1 set of drifts or expanders for boiler tubes.

1 set of safety-valve springs (if so fitted) for every four valves; if there are not four valves, then at least one set of springs must be carried.
1 screw jack.
And a set of engineer's tools suitable for the service, including hammers and chisels for vice and forge; solder and soldering-iron; sheets of tin and copper; spelter; muriatic acid, or other equivalent, &c., &c.

LLOYD'S RULES.

Lloyd's Requirements as to Spare Gear are as follows:—The articles of spare gear mentioned in the following list will be required to be carried in all steam vessels classed in the Society's Register Book, viz.:—

2 connecting-rod, or piston-rod, top end bolts and nuts.
2 „ bottom end bolts and nuts.
2 main bearing bolts.
1 set of coupling bolts.
1 „ feed- and bilge-pump valves.
1 „ piston springs (where common springs are used).
A quantity of assorted bolts and nuts.
Iron of various sizes.

In addition to the foregoing, the following articles are recommended to be carried, with a view to expedite repairs and lessen delay in distant ports:—

Crank shaft.
Propeller shaft.
„ or full set of blades.
Stern bush, or lignum vitæ lining in bush.
1 pair connecting-rod brasses.
1 „ crosshead brasses.
1 set of check valves.
6 cylinder cover bolts.
6 junk ring bolts.
4 valve chest bolts.
1 set of link brasses.
1 eccentric strap, complete.
1 air-pump rod.
1 circulating pump rod.
H.P. valve spindle.
L.P „
2 dozen boiler tubes.
3 „ condenser tubes.
1 cylinder escape valve spring.
1 set of safety valve springs.

BRITISH CORPORATION RULES.

2 connecting-rod top end bolts.
2 „ bottom end bolts.
2 main bearing bolts.
1 set of coupling bolts.
1 „ feed- and bilge-pump valves.

1 set of piston springs, if common springs are used.
1 set of safety valve springs.
6 cylinder cover studs and nuts.
6 valve chest cover studs and nuts.
An assortment of bolts, studs, and nuts.
Bar and plate iron, assorted.
½ set of fire bars.

Bureau Veritas Rules.

Note.—For navigation letter G, no other article than those marked with an asterisk shall be required.

1 propeller or ½ set of blades.
For twin screws with detachable blades, 1 blade for each screw.
1 slide valve spindle suitable for each engine.
1 pair of crank-pin brasses, suitable for each engine.
1 ,, crosshead brasses.

(If the brasses are lined with anti-friction metal, a suitable quantity of this metal, with the necessary tools for lining the brasses, will be accepted instead of the spare brasses.)

1 set of slide blocks for valve link motion.
1 eccentric strap or brass liner for same.
*2 connecting-rod bolts and 2 crosshead bolts.
*1 set of coupling bolts for 1 coupling.
1 circulating and 1 air-pump rod.
1 complete set of valves for circulating and air-pumps.
*½ set feed-pump valves.
½ ,, of boiler feed valves.
½ ,, bilge-pump valves.
½ ,, safety valve springs.
2 per cent. of the number of condenser tubes.
5 ,, ,, boiler ,,
*½ set of grate bars.
2 sets of gauge glasses.

The spare propeller must be secured on a deck by angle iron lugs riveted through the deck and beams, or other efficient means.

It is recommended to provide the same spare gear for the donkey engines and other auxiliary engines as for the main engines.

When the circulating pump is of the rotary system with a separate engine, the pump rod mentioned above is to be replaced by the following articles:—

1 steam piston and piston-rod.
1 slide valve and spindle.
1 set of brasses.

§ 2. The following articles are recommended for vessels classed for Atlantic and long voyages, to facilitate repairs in distant ports:—

1 crank shaft, when the crank shaft is in one piece, or
1 element of a shaft. when it is made of several interchangeable parts.
1 complete piston } for each cylinder.
1 cylinder cover }

§ 3. Vessels provided with all articles mentioned in §§ 1 and 2 above will be entered in the register with a special mark ○ in the third column of the page where particulars of machinery are described.

§ 4. All vessels to have a spare tiller and relieving tackles.

Steam vessels to be provided with the tools required for the working and overhauling of the propelling and other machinery.

ADMIRALTY SPARE GEAR FOR THE MAIN MACHINERY OF A CRUISER.

Bolts and nuts complete for main shaft couplings,	$\frac{1}{2}$ set for 1 coupling of each pattern
Bolts and nuts for main bearings, complete,	1 ,, 1 main bearing ,,
Bolts and nuts turned and fitted for engines, in addition to those specified elsewhere, properly assorted,	100.
Brasses, top and eccentric-rods, with liners, complete,	2 sets for 1 rod of each pattern.
Connecting-rod, bolts, and brasses, with liners, complete for each end,	2 sets of brasses for 1 rod of each pattern, and 1 set bolts each pattern.
Copper expansion rings for L.P. slide relief rings (if fitted),	1 for each fitted.
Cylinder faces or piston valve liners, with studs or nuts or screws to secure each,	1 of each pattern.
Eccentric-rod and strap, with bolts, nuts, liners, &c., complete,	1 number of each pattern.
Filtering material and frames for main engine grease extractors,	1 set of frames and 6 sets of material for each extractor.
Gear for removing or replacing metallic packing,	1 set for each size of rod or packing.
Glasses for telegraphs, tell-tales, and revolution telegraphs, properly marked and coloured,	,, pedestal or dial.
Gunmetal sectors for link blocks, complete,	,, set of valve gear for 1 set of engines.
Hoses, suction, No. 4 gauge, sufficient to enable any crank-pit to be connected to the bilge suction direct-box in engine-room, with gunmetal end,	1 in number, complete, for each engine-room.
Lignum vitæ strips and facings for all parts, so fitted throughout and allowance made for final setting,	1 set for each bearing, so fitted for 1 set of engines.
Packing metallic of each size,	Sufficient for 2 piston-rods and for 2 other rods of each pattern so fitted.
Packing rings, screws, and fittings, complete, for flat slide-valves,	1 set for 1 valve of each pattern.
Packing rings and springs, complete, with bolts or studs and nuts for pistons, piston valves, and balance pistons,	1 set for 1 H.P., 1 set for 1 I.P., and 2 sets for 1 L.P. piston. 1 set for each end for 1 H.P. and 1 I.P. piston slide valve. 2 sets for 1 L.P. balance piston. 1 set for 1 H.P. and 1 I.P. balance piston (if so fitted).
Piston, complete for L.P. cylinder,	1 number.
Piston-rod crosshead slipper, complete,	1 number of each pattern and size, if not supplied with the piston-rod.
Propeller bolts, covering plates, studs, &c.,	1 set, complete, for 1 blade of each hand.
Screwed glands for tubes of condensers,	$\frac{1}{30}$ of the whole number fitted.
Shackles for propeller blade,	2 numbers.
Slide-rods, with brasses, nuts, &c., complete,	1 number of each size.
Split pins, special, as required,	2 of each kind.
Springs, excepting when otherwise specified,	1 for each 4 or less number of each description fitted.
Steel rods or plates as protective fittings for condensers,	3 sets.
T pieces for deck plates of fire service (portable or fixed),	1 for each pattern.
Thrust collar,	1 set for 1 set of engines.

Tools for tubing surface condensers, - - -	1 set, complete, of Admiralty pattern comprising:—1 die nut for screwed glands; 1 hollow tap to fit the screwed holes in tube-plate when tube is in place; 2 hollow punches, with solid ends, for packing glands; 4 wrenches, L-handled, for screwed gland; 4 wrenches, T-handled, for screwed gland; 2 drifts, with collars, for driving out tubes; 2 drifts, taper, for tightening tubes. All these tools to be clearly marked "Condensers."
Valves, india-rubber, - - - - - - -	2 numbers for each fitted.
Valves, metallic, for pumps, with bolts or studs, nuts, and guards, ready for facing, complete,	1 number for each 2 fitted.
White metal bearings, mandrels for, - - -	1 set.

SPARE GEAR FOR BOILERS (ADMIRALTY).

Description.	Cylindrical.	Yarrow.
Automatic feed-box and apparatus, complete, with pipes and valves up to boilers, - - - - -	...	2 numbers of each kind.
Bearer bars for one boiler, - -	4 sets.	4 sets, or 1 set for each boiler of different width.
Blank flanges for testing and other purposes, - - - - - -	1 complete set.	1 complete set.
Bolts and nuts for other parts (exclusive of casings, - - - -	...	⅓ total number of each size fitted.
Doors' examination, - - - - -	⅙ total number fitted.	⅙ total number fitted.
Tubes, - - - - - - -	$\frac{1}{20}$,, ,,	$\frac{1}{20}$,, ,,
Furnace bars, - - - - - -	¼ set.	¼ set for each boiler.
Hoses and branch pipes for fire-extinguishing and ash-pan water service. - - - - - - -	1 complete set.	1 complete set.
Hoses for emptying boilers, - -	,,	,,
Packing glands, hexagonal, metallic, for both ends of water-gauge mountings, including those for separators,	¼ total number of each size fitted.	¼ total number of each size fitted.
Springs for safety- and escape-valves,	2 sets of each size.	2 sets of each size.
Smoke-directing plates or bafflers, -	...	Sufficient for 1 boiler of each size.
Short screwed stays for boilers, $\frac{1}{16}$ inch larger than required when new,	Sufficient for 1 boiler.	...
Valves, feed, non-returning, without boxes, - - - - - - -	¼ total number fitted.	¼ total number fitted.
Washers and other parts of boilers not separately specified above, excepting steam collectors, casings, and brickwork, - - - - -	Sufficient for 1 boiler.	Sufficient for 1 boiler.
Cap ferrules, - - - - - -	1 complete set.	...
Retarders, - - - - - -	,,	...

NOTE.—The internal parts of all the boilers are to be thoroughly dried and the ends then plugged or closed, and the fireside, excepting where directed, coated with linseed oil.

ERS CALIFORN

SPARE GEAR FOR AUXILIARY MACHINERY (ADMIRALTY).

Brasses, main bearings as fitted.

„ „ for each fitted.

„ connecting-rod, 1 set, complete, for each fitted for each end.

Eccentric-rod and strap, with bolts, nuts, and pin, complete, or equivalent fittings for each fitted.

Packing rings for piston, and piston valves for each fitted.

„ water pistons for each fitted.

Slide or piston valves, complete, for each fitted.

Springs for pistons and all other parts for each fitted.

Steam actuating and distributing valves and chests, complete (for direct-acting pumps) of each pattern for each 2 fitted.

Slide motion rods or gear, complete (for direct-acting pumps) of each pattern for each 2 fitted.

Brasses or bushes for all other working parts, pair or number, for each fitted.

Piston-rod with nuts, complete, with crosshead and slipper of each pattern fitted.

Slide-rod with nuts, complete, of each pattern fitted.

Metallic packing for glands, with withdrawing gear, for each set fitted.

Valves, metallic, with bolts or studs, nuts, and guards, complete, for each 2 fitted.

Valves, india-rubber, if so fitted, for each.

Pump-rods with nuts, complete.

Brasses, bushes or facings, for pump shaft, with lignum vitæ, complete, for 1 pump.

Wedges, for each 4 fitted.

Distilling condenser coils or tubes, suitable for any condenser, for 1 condenser.

Evaporator coils or tubes and tube plates or connections, complete, ready for immediate fitting in place, suitable for any evaporator, for 1 evaporator.

Evaporator coils or tubes, separate, of each pattern fitted.

Glasses for pedestals and dials, properly coloured and marked, for each set fitted.

All spanners and tools to be plainly marked.

APPENDIX D.

Board of Trade Rules for Boilers.

The Surveyor's Duty and Responsibility in Fixing Pressures.—The Surveyor is required by the Act to fix the limits of weight to be placed on the safety-valves of passenger steamships. The Surveyor having himself fixed the limits of the weight, is then required to declare, that in his judgment the boiler and machinery are sufficient for the service intended, and in good condition, and that they will be sufficient for twelve months, or such other period as he may, in his judgment, determine. For his guidance the following suggestions are given:—

Working Pressure to be Fixed by Calculation, &c.—The Surveyor should fix the working pressure for boilers by a series of calculations of the strength of the various parts, and according to the workmanship and material. The senior Engineer Surveyors should receive and report on any plans of boilers intended for passenger vessels that may be submitted in due course with the Form Surveys 6. They are not to report on any tracing or plan that is not accompanied by that form. When the Surveyor has received plans and tracings of new boilers, or of alterations of boilers, and has approved of them, he will of course be careful in making his examinations from time to time to see that they are followed in construction. When he has not had the plans submitted, but is called in to survey a boiler, he will then measure the parts, note the details of construction, and if necessary bore the plates to ascertain their thickness, &c., before he gives his declaration. And in the event of any novelty in construction, or of any departure from the practice of staying and strengthening noted in these regulations, he should report full particulars to the Board of Trade before fixing the working pressure.

The Surveyor cannot declare a boiler to be safe of whose construction, material, and workmanship he is not fully informed. He should, therefore, be very careful how he ventures to give a declaration for a boiler that he is not called in to survey until after it is completed and fixed in the ship.

Girders for Flat Surfaces.—When the tops of combustion boxes, or other parts of a boiler, are supported by solid rectangular girders, the following formula, which is used by the Board of Trade, will be useful for finding the working pressure to be allowed on the girders, assuming that they are not subjected to a greater temperature than the ordinary heat of steam, and in the case of combustion chambers, that the ends are properly bedded to the edges of the tube plate and the back plate of the combustion box:—

$$\frac{C \times d^2 \times T}{(W - P) D \times L} = \text{Working pressure.}$$

W = Width of combustion box in inches.
P = Pitch of supporting bolts in inches.
D = Distance between the girders from centre to centre in inches.
L = Length of girder in feet.
d = Depth of girder in inches.
T = Thickness of girder in inches.
N = Number of supporting bolts.
$C = \frac{N \times 1{,}320}{N + 1}$ when the number of bolts is odd.
$C = \frac{(N + 1)\, 1{,}320}{N + 2}$ when the number of bolts is even.

The working pressure for the supporting bolts, and for the plate between them, shall be determined by the rule for ordinary stays and plates.

Plates for Flat Surfaces.—The pressure on plates forming flat surfaces will be easily found by the following formula:—

$$\frac{C \times (T + 1)^2}{S - 6} = \text{Working pressure.}$$

T = Thickness of the plate in sixteenths of an inch.
S = Surface supported in square inches.
C = Constant according to the following circumstances:—

C = 240 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of plates and outside doubling-strips of the same thickness as the plates they cover, and not less in width than two-thirds of the pitch of the stays.

NOTE.—When doubling-plates cover the whole of the flat surface, the case should be submitted for the consideration of the Board.

C = 210 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of plates and washers, the latter on the outside of boiler, being at least two-thirds the pitch of the stays in diameter, and the same thickness as the plates they cover. These washers to be riveted on the plate.

C = 165 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of plates and outside washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plates they cover.

C = 150 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts only on both sides of plates.

C = 75 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, and the stays fitted with nuts on both sides of plates, and outside washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plate they cover.

C = 67·5 when the plates are exposed to the impact of heat or flame, and steam in contact with the plate, and the stays fitted with nuts only on both sides of plates.

C = 100 when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plate and fitted with nuts.

C = 66 when the plates are exposed to the impact of heat or flame, with water in contact with the plate, and the stays screwed into the plate, having the ends riveted over to form substantial heads.

C = 39·6 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, with the stays screwed into the plate, and having the ends riveted over to form substantial heads.

When a circular flat end is bolted or riveted to a cylindrical shell, S in the formula may be taken as the area of the square inscribed in the circle passing through the centres of the bolts or rivets securing the end, provided the angle ring or flange is of sufficient thickness.

Compressive Stress on Steel Tube Plates.—

$$\frac{(D - d)\,T \times 28{,}000}{W \times D} = \text{Working pressure.}$$

D = Least horizontal distance between centres of tubes in inches.

d = Inside diameter of ordinary tube in inches.

T = Thickness of tube plate in inches.

W = Extreme width of combustion box in inches from front of tube plate to back of fire box, or distance between combustion box tube plates when boiler is double-ended and the box common to the furnaces at both ends.

Cylindrical Boilers.—When cylindrical boilers are made of the best material, with all the rivet holes drilled in place, and all the seams fitted with double butt straps, each of at least five-eighths the thickness of the plates they cover, and all the seams at least double-riveted with rivets having an allowance of not more than 75 per cent. over the single shear, and provided that the boilers have been open to inspection during the whole period of construction, then 4·5 may be used as the factor of safety. The tensile strength of the material is to be taken as equal to 27 tons per square inch when mild steel is used, tested in accordance with the Rules (p. 442). When plates of higher tensile strength than 27 tons are used, the actual strength, as found on test, may be substituted in the following Rule. The boilers must be tested by hydraulic pressure to twice the working pressure in the presence and to the satisfaction of the Surveyor:—

To ascertain the strength of shell, the relative sectional areas of plate and rivet must first be determined by the following formulæ:—

$$\frac{(\text{Pitch} - \text{dia. of rivet}) \times 100}{\text{Pitch}} = \left\{\begin{array}{l}\text{Percentage of strength of plate at joint}\\ \text{as compared with solid plate.}\end{array}\right.$$

For maximum permissible pitch of rivets, &c., see formulæ and sketches at end of this section. (Pp. 277 to 281, Seaton and Rounthwaite's *Pocket-Book of Marine Engineering.*)

$$\frac{(\text{Area of rivet} \times \text{No. of rows of rivets}) \times 100}{\text{Pitch} \times \text{thickness of plate}} = \left\{\begin{array}{l}\text{Percentage of strength}\\ \text{of rivets as compared}\\ \text{with solid plate.}\end{array}\right.$$

If the rivets are in double shear, multiply the percentage thus found by 1·75.

In consequence of the low shearing strength of steel rivets, the Board require that in all types of joint the *nominal* rivet section shall be not less than $\frac{27}{23}$ of the net plate section; thus, in order that rivet section may be considered to have the same strength as plate section their relation must be:—

In lap joints—

$$\begin{aligned}&(\text{Area of rivet} \times \text{No. of rows of rivets}) \times 23\\ &= (\text{Pitch} - \text{diameter of rivet}) \times \text{thickness of plate} \times 27.\end{aligned}$$

And in butt joints—

$$\begin{aligned}&(\text{Area of rivet} \times \text{No. of rows of rivets} \times 1{\cdot}75) \times 23\\ &= (\text{Pitch} - \text{diameter of rivets}) \times \text{thickness of plate} \times 27.\end{aligned}$$

But when plate of a higher minimum tensile strength than 27 tons is used, the actual strength in tons should be used instead of the multiplier 27.

The working pressure per square inch that may be allowed on the safety valves is then given by—

$$\text{Working pressure} = \frac{S \times \% \times 2\,T}{D \times F},$$

Where S = Tensile strength of material in lbs. per square inch.
$\%$ = One of the two percentages, found by Rules above divided by 100.
T = Thickness of plate in inches.
D = Inside diameter of boiler in inches (inside diameter of outer strake, if any).
F = Factor of safety from following Table if $\%$ refers to plate section.
F = 4·5 if $\%$ refers to rivet section.

The smaller of the two results to be taken.

Various penal additions are provided in the rules for cases in which the workmanship is of inferior character.

If it is proposed to use steel for superheaters, particulars should be submitted to the Board for consideration.

TABLE LXIX.—Board of Trade Factors of Safety.

When all rivet holes are drilled in place after bending; *all* seams fitted with double butt-straps, each at least five-eighths the thickness of the plates they cover; all seams at least double riveted; and boilers open to inspection during construction, -	F = 4·5
To be added when circumferential seams are lap and double riveted, -	·1
To be added when longitudinal seams are lap and double riveted, -	·2
To be added when longitudinal seams are lap and treble riveted, -	·1
To be added when boiler is of such length as to fire from both ends, unless middle circumferential seams are lap and treble riveted, -	·3

Compensating rings should be fitted around all manholes and openings, of at least the same effective sectional area as the plate cut out, and in no case should rings be of less thickness than plate to which they are attached. Manholes in shells of cylindrical boilers should have their shorter axis placed longitudinally.

It is very desirable that compensating rings round openings in flat surfaces be of L or T iron.

The neutral parts of shells under steam domes must be efficiently stiffened and stayed.

Board of Trade Rules for Cylindrical Boiler Shells.

JOINTS WITH DRILLED HOLES.

Formulæ for ordinary chain riveted and ordinary zig-zag riveted joints, and for joints of these descriptions, when every alternate rivet in the outer or in the outer and inner rows have been omitted:—

Let E = distance from edge of plate to centre of rivet in inches.
V = distance between rows of rivets in inches.
V_1 = distance between inner and middle row of rivets in inches, for chain and zig-zag treble riveted butt joints with alternate rivets omitted in outer rows.
B = boiler pressure in lbs. per square inch.
C = 1 for lap or single butt joints.
C = 1·75 for double butt joints.
d = diameter of rivets in inches.
D = inside diameter of boiler in inches.
F = factor of safety for shell plates.
n = number of rivets in one pitch.
p_D = diagonal pitch in inches.
P_D = diagonal pitch in inches between inner and middle rows of rivets in inches, for zig-zag treble riveted butt joints with alternate rivets omitted in outer rows
p = greatest pitch of rivets in inches.
r = percentage of plate left between holes in greatest pitch.
R = percentage of rivet section.
R_1 = percentage of combined plate and rivet section.
S = tensile strength of material in lbs. per square inch of section.
S_1 = tensile strength of plates in tons.
T = thickness of plate in inches.
T_1 = thickness of each butt strap in inches.
% = least value of r, R, or r, R, R_1, as the case may be, divided by 100.

ORDINARY CHAIN AND ZIG-ZAG RIVETED JOINTS.

Iron plates and iron rivets, or steel plates and steel rivets:—

$$\frac{100\,(p - d)}{p} = r.$$

Steel plates and steel rivets:—

$$\frac{100 \times 23 \times d^2 \times {\cdot}7854 \times n \times C \times F}{4{\cdot}5 \times S_1 \times p \times T} = R.$$

Given C, d, F, n, T, to find p, so that r and R are equal.

Steel plates and steel rivets:—

$$\frac{23 \times d^2 \times {\cdot}7854 \times n \times C \times F}{4{\cdot}5 \times S_1 \times T} + d = p.$$

Given C, F, n, T, r, to find p and d.

Steel plates and steel rivets:—

$$\frac{4{\cdot}5 \times S_1 \times r \times T}{23 \times (100 - r) \times {\cdot}7854 \times n \times C \times F} = d.$$

$$\frac{100 \times 4{\cdot}5 \times S_1 \times r \times T}{23 \times (100 - r)^2 \times {\cdot}7854 \times n \times C \times F} = p.$$

Steel plates and steel rivets when d is found first, then :—

$$\frac{100\,d}{100 - r} = p.$$

Steel plates and steel butt straps.

Double butt straps :—

$$\frac{5 \times T}{8} = T_1.$$

Single butt straps :—

$$\frac{9 \times T}{8} = T_1.$$

For Distance between Rows of Rivets, &c., for Double, Treble, and Quadruple Riveted Lap Joints; and Double and Treble Riveted Butt Joints with same Number of Rivets in Inner and in Outer Rows.

Steel plates and steel rivets :—

$$\frac{3 \times d}{2} = E.$$

Chain-riveted joints, not less than :—

$$2 \times d = V.$$

Zig-zag riveted joints :—

$$\frac{\sqrt{(11\,p + 4\,d)(p + 4\,d)}}{10} = V.$$

Diagonal pitch for zig-zag riveted joints :—

$$\frac{6\,p + 4\,d}{10} = p_D.$$

To determine the Working Pressure.

$$\frac{S \times \% \times 2\,T}{F \times D} = B.$$

Chain and Zig-zag Riveted Joints in which every alternate Rivet has been omitted in the Outer Row, or in the Outer and the Inner Rows.

Steel plates and steel rivets :—

$$\frac{100\,(p - d)}{p} = r.$$

Steel plates and steel rivets :—

$$\frac{100 \times 23 \times d^2 \times \cdot 7854 \times n \times C \times F}{4\cdot 5 \times S_1 \times p \times T.} = R.$$

Steel plates and steel rivets :—

$$\frac{100\,(p - 2\,d)}{p} + \frac{R}{n} = R_1.$$

When the Joints are fitted with Single or Double Butt Straps and the Number of Rivets in the Inner Row is double the Number in the Outer Row.

Steel plates and steel butt straps.

Double butt straps :—

$$\frac{5 \times T\,(p - d)}{8 \times (p - 2\,d)} = T_1.$$

Single butt straps :—

$$\frac{9 \times T(p - d)}{8 \times (p - 2d)} = T_1.$$

When the number of rivets in the inner row is the same as in the outer row.

Double butt straps :—

$$\frac{5 \times T}{8} = T_1.$$

Single butt straps :—

$$\frac{9 \times T}{8} = T_1.$$

For Distance between Rows of Rivets, &c., for Double and Treble Riveted Butt Joints with Alternate Rivets omitted in Outer Rows; and Treble Riveted Lap Joints and Butt Joints with alternate Rivets omitted in Outer and in Inner Rows.

Steel plates and steel rivets :—

$$\frac{3 \times d}{2} = E.$$

Chain-riveted joints :—

$$\left.\begin{array}{l} \dfrac{\sqrt{(11p + 4d)(p + 4d)}}{10} = V, \\ \text{or} \qquad 2 \times d = V. \end{array}\right\} \text{The greatest of these two values of V to be used. } \textit{See Note below.}$$

For treble chain-riveted joint with alternate rivets omitted in outer row :—

$$2 \times d = V_1.$$

Zig-zag riveted joints :—

$$\sqrt{(\tfrac{11}{20}p + d)(\tfrac{1}{20}p + d)} = V.$$

Diagonal pitch :—

$$\tfrac{3}{10}p + d = p_D.$$

For treble zig-zag riveted joint with alternate rivets omitted in outer row :—

$$\frac{\sqrt{(11p + 8d)(p + 8d)}}{20} = V_1.$$

Diagonal pitch :—

$$\frac{3p + 4d}{10} = P_D.$$

To determine the Working Pressure.

$$\frac{S \times {}^{\circ}\!/\!_{\circ} \times 2T}{F \times D} = B.$$

NOTE.—The minimum value of V or V_1 for chain-riveted joints is given as $2d$, but $\frac{4d + 1}{2}$ is more desirable.

Maximum Pitches for Riveted Joints.

T = thickness of plate in inches.
p = maximum pitch of rivets in inches, provided it does not exceed 10 inches.
C = constant applicable from the following table :—

TABLE LXX.

Number of Rivets in One Pitch.	Constants for Lap Joints.	Constants for Double-Butt Strap Joints.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	...	6·00

$$(C \times T) + 1\tfrac{5}{8} = p.$$

When the work is first-class, such pitches may be adopted so far as safety is concerned; yet, in some cases, it may be well not to adopt the greatest pitch found by the formula. The maximum should *not*, however, exceed 10 inches with the thickest plates for boiler shells.

If, in any case, the pitch is found to exceed that arrived at by the foregoing formula, for the particular description of joint and thickness of plate, such pitches should *not* be passed, but, in all cases, reported.

APPENDIX E.

LLOYD'S RULES FOR DETERMINING THE WORKING PRESSURE TO BE ALLOWED IN NEW BOILERS.

CYLINDRICAL SHELLS OF STEEL BOILERS.

THE strength of cylindrical shells of steel boilers is to be calculated from the following formula:—

$$\frac{C \times (T - 2) \times B}{D} = \text{working pressure in lbs. per square inch.}$$

Where D = mean diameter of shell in inches.
T = thickness of plate in sixteenths of an inch.
C = 21, when the longitudinal seams are fitted with double butt straps of equal width,
C = 20·25, when they are fitted with double butt straps of unequal width, only covering on one side the reduced section of plate at the outer lines of rivets.
C = 19·5, when the longitudinal seams are lap joints.
If the minimum tensile strength of shell plates is 28 or 29 tons per square inch, instead of 27 tons per square inch, these values of C may be correspondingly increased.
B = the least percentage of strength of longitudinal joint,* found as follows:—

For plate at joint, $B = \dfrac{p - d}{p} \times 100.$

„ rivets „ $B = \dfrac{n \times a}{p \times t} \times 85$, where steel rivets are used.

$B = \dfrac{n \times a}{p \times t} \times 70$, „ iron „

Where p = pitch of rivets in inches.
t = thickness of plate in inches.
d = diameter of rivet holes in inches.
n = number of rivets used per pitch in the longitudinal joint.
a = sectional area of rivet in square inches. In case of rivets in double shear $1{\cdot}75\,a$ is to be used instead of a.

NOTE.—For the shell plates of superheaters or steam chests enclosed in the uptakes or exposed to the direct action of the flame, the coefficients should be ⅔ of those given in the preceding tables.

Proper deductions are to be made for openings in shell.

All manholes in circular shells to be stiffened with compensating rings.

The shell plates under domes in boilers so fitted to be stayed from the top of the dome or otherwise stiffened.

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* The inside butt strap to be at least ¾ of the strength of the longitudinal joint.

STAYS.

The strength of stays supporting flat surfaces is to be calculated from the smallest part of the stay or fastening, and the strain upon them is not to exceed the following limits, namely :—

Iron Stays.—For stays not exceeding 1½ inches smallest diameter, and for all stays which are welded, 6000 lbs. per square inch; for unwelded stays above 1½ inches smallest diameter, 7500 lbs. per square inch.

Steel Stays.— For *screw* stays not exceeding 1½ inches smallest diameter, 8000 lbs. per square inch; for *screw* stays above 1½ inches smallest diameter, 9000 lbs. per square inch. For other stays not exceeding 1½ inches smallest diameter, 9000 lbs. per square inch, and for stays exceeding 1½ inches smallest diameter, 10,000 lbs. per square inch. No steel stays are to be welded.

Stay Tubes.—The stress is not to exceed 7500 lbs per square inch.

FLAT PLATES.

The strength of flat plates supported by stays is to be taken from the following formula :—

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

Where T = thickness of plate in sixteenths of an inch.
P^2 = square of pitch in inches. If the pitch in the rows is not equal to that between the rows, then the mean of the squares of the two pitches is to be taken.
C = 90 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with screw stays with riveted heads.
C = 100 for iron or steel plates above $\frac{7}{16}$ thick, fitted with screw stays with riveted heads.
C = 110 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with stays and nuts.
C = 120 for iron plates above $\frac{7}{16}$ thick, and for steel plates above $\frac{7}{16}$ and under $\frac{9}{16}$ thick, fitted with screw stays and nuts.
C = 135 for steel plates $\frac{9}{16}$ thick and above, fitted with screw stays and nuts.
C = 140 for iron plates, fitted with stays with double nuts.
C = 150 for iron plates, fitted with stays with double nuts and washers outside the plates, of at least ⅓ of the pitch in diameter and ½ the thickness of the plates.
C = 160 for iron plates, fitted with stays with double nuts and washers riveted to the outside of the plates, of at least ⅖ of the pitch in diameter and ½ the thickness of the plates.
C = 175 for iron plates, fitted with stays with double nuts and washers riveted to the outside of the plates, when the washers are at least ⅔ of the pitch in diameter and of the same thickness as the plates.

For iron plates fitted with stays with double nuts and doubling strips riveted to the outside of the plates, of the same thickness as the plates, and of a width equal to ⅔ the distance between the rows of stays, C may be taken as 175, if P is taken to be the distance between the rows, and 190 when P is taken to be the pitch between the stays in the rows.

For steel plates, other than those in combustion chambers, the values of C may be increased as follows :—

C = 140 increased to 175.
C = 150 „ 185.
C = 160 „ 200.
C = 175 „ 220.
C = 190 „ 240.

If flat plates are strengthened with doubling plates securely riveted to them, having a thickness of not less than $\frac{2}{3}$ of that of the plates, the strength to be taken from

$$\frac{C \times \left(T + \frac{t}{2}\right)^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

Where T = thickness of doubling plates in sixteenths, and C, T, and P are as above.

NOTE.—In the case of front plates of boilers in the steam space, these numbers should be reduced 20 per cent., unless the plates are guarded from the direct action of the heat.

For steel tube plates in the nest of tubes the strength to be taken from

$$\frac{140 \times T^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

Where T = the thickness of the plates in sixteenths of an inch.
P = the *mean* pitch of stay tubes from centre to centre.

For the wide water spaces between the nests of tubes, the strength to be taken from

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

Where P = the horizontal distance from centre to centre of the bounding rows of tubes; and
C = 120, where the stay tubes are pitched with two plain tubes between them and are not fitted with nuts outside the plates.
C = 130, if they are fitted with nuts outside the plates.
C = 140, if each alternate tube is a stay tube not fitted with nuts.
C = 150, if they are fitted with nuts outside the plates.
C = 160, if every tube in these rows is a stay tube and not fitted with nuts.
C = 170, if every tube in these rows is a stay tube and each alternate stay tube is fitted with nuts outside the plates.

The thickness of tube plates of Combustion Chambers, in cases where the pressure on the top of the chambers is borne by these plates, is not to be less than that given by the following rule :—

$$T = \frac{P \times W \times D}{1750 \times (D - d)}.$$

Where P = working pressure in lbs. per square inch.
W = width of Combustion Chamber between plates in inches.
D = horizontal pitch of tubes in inches.
d = inside diameter of plain tubes in inches.
T = thickness of tube plates in sixteenths of an inch.

GIRDERS.

The strength of girders supporting the tops of combustion chambers and other flat surfaces is to be taken from the following formula:—

$$\frac{C \times d^2 \times T}{(L - P) \times D \times L} = \text{working pressure in lbs. per square inch.}$$

Where L = width between tube plates, or tube plate and back plate of chamber.
P = pitch of stays in girders.
D = distance from centre to centre of girders.
d = depth of girder at centre.
T = thickness of girder at centre.
All these dimensions to be taken in inches.

Wrought Iron.

C = 6,000, if there is one stay to each girder
C = 9,000, if there are two or three stays to each girder.
C = 10,000, if there are four or five stays to each girder.
C = 10,500, if there are six or seven stays to each girder.
C = 10,800, if there are eight stays or above to each girder.

Wrought Steel.

C = 6,600, if there is one stay to each girder.
C = 9,900, if there are two or three stays to each girder.
C = 11,000, if there are four or five stays to each girder.
C = 11,550, if there are six or seven stays to each girder.
C = 11,880, if there are eight stays or above to each girder.

CIRCULAR FURNACES.

The strength of plain furnaces to resist collapsing to be calculated as follows:—

Where the length of the plain cylindrical part of the furnace exceeds 120 times the thickness of the plate, the working pressure is to be calculated by the following formula:—

$$\frac{1{,}075{,}200 \times T^2}{L \times D} = \text{working pressure in lbs. per square inch.}$$

Where the length of the plain cylindrical part of the furnace is less than 120 times the thickness of the plate, the working pressure is to be calculated by the following formula:—

$$\frac{50 \times (300\,T - L)}{D} = \text{working pressure in lbs. per square inch.}$$

Where D = outside diameter of furnace in inches.
T = thickness of plates in inches.
L = length of plain cylindrical part in inches, measured from the centres of the rivets connecting the furnaces to the flanges of the end and tube plates, or from the commencement of the curvature of the flanges of the furnace where it is flanged or fitted with Adamson rings.

In the furnaces referred to below, the formulæ given are applicable if the steel used has a tensile strength of not less than 26 nor more than 30 tons per square inch. If the material of furnaces has a less tensile strength than 26 tons per square inch, then, for each ton per square inch which the minimum tensile strength falls below 26, the coefficient is to be correspondingly decreased by $\frac{1}{26}$ part.

The strength of corrugated furnaces made of steel, on Fox's, Morison's, Deighton's, or Beardmore's plan, and of the Leeds Forge bulb furnace, to be calculated from

$$\frac{1259 \times (T - 2)}{D} = \text{working pressure in lbs. per square inch.}$$

The strength of Brown's cambered, and improved Purves' furnaces (with ribs 9 inches apart) to be calculated from the following formula:—

$$\frac{1160 \times (T - 2)}{D} = \text{working pressure in lbs. per square inch.}$$

The strength of spirally corrugated furnaces is to be calculated from the following formula:—

$$\frac{912 \times (T - 2)}{D} = \text{working pressure in lbs. per square inch.}$$

Where T = thickness of plate in sixteenths of an inch; and
D = outside diameter of corrugated furnaces, in inches; or *smallest* outside diameter, in inches, of Brown's cambered, improved Purves', and Leeds Forge bulb furnaces.

The strength of Holmes' patent furnaces, in which the corrugations are not more than 16 inches apart from centre to centre, and not less than 2 inches high, to be calculated from the following formula:—

$$\text{Working pressure in lbs. per square inch} = \frac{945 \times (T - 2)}{D}.$$

Where T = thickness of plain portions of furnace in sixteenths of an inch.
D = outside diameter of plain parts of the furnace in inches.

APPENDIX F.

RULES OF THE BRITISH CORPORATION FOR THE WORKING PRESSURE, THICKNESSES OF PLATES, AND SIZES OF STAYS FOR STEEL BOILERS.

THE rules following are intended to apply to the construction of steel boilers; where boilers are to be made of iron, they will be specially considered by the committee.

Strength Calculations.—The sizes and arrangement of the different parts for a given working pressure, or the working pressure suitable to a given size and arrangement of material, may be found from the following formulæ and rules :—

Cylindrical Shells—

$$\frac{C \times (T - 1) \times E}{D} = W.$$

Where C = 20·4, when the longitudinal seams are fitted with double butt straps of equal width, and of thickness at least equal to that obtained from the formula for butt straps.

C = 19·7, when the double butt straps are of unequal width—*i.e.*, one strap not covering the outer row of rivets, with the thickness as before.

C = 19·0, when the longitudinal seams are lap joints.

T = thickness of shell plate in sixteenths of an inch.

E = the least percentage of strength of longitudinal joints, found as follows :—

$$\text{For the plate at the joint, } E = \frac{p - d}{p} \times 100.$$

$$\text{For the rivets at the joint, } E = \frac{n \times a}{p \times t} \times \begin{cases} 85 \text{ for steel rivets.} \\ 70 \text{ for iron rivets.} \end{cases}$$

Where p = pitch of rivets in inches.

d = diameter of rivet holes in inches.

n = number of rivets used per pitch.

a = sectional area of rivets in square inches.

t = thickness of plate in inches.

Where rivets are in double shear, 1·75 a is to be used instead of a.

D = mean diameter of shell in inches.

W = working pressure in lbs. per square inch.

NOTE.—The constant, C, is to be used with steel of the minimum strength of 28 tons per square inch, as required by the rules (see p. 588); when a higher minimum strength is guaranteed, the constant may be proportionately increased.

Double Butt Straps.—When all the rows of rivets in the butt are of the same pitch, each strap must be at least five-eighths of the thickness of the shell plate, and, when the pitch of the outer row of rivets is twice that of the centre row, the thickness of each plate is to be in accordance with the following formula:—

$$T_1 = T\frac{5(p-d)}{8(p-2d)}.$$

Where T_1 = thickness of strap in sixteenths of an inch.
T = thickness of shell in sixteenths of an inch.
p = pitch of rivets in inches.
d = diameter of rivet holes in inches.

Flat Surfaces, supported by Stays—

$$\frac{C \times (T-1)^2}{P^2 + p^2} = W.$$

Where C = 250 for plates fitted with screwed stays having riveted heads.
C = 330 for plates fitted with screwed stays and nuts.
C = 400 for plates fitted with stays and double nuts.
C = 436 for plates with stays, double nuts, and washers outside. The washers to be at least half the thickness of the plate, and in diameter equal to one-third the pitch of stays.
C = 476 for plates fitted with stays, double nuts, and washers outside riveted to the plate. The washers to be at least half the thickness of the plate, and in diameter equal to two-fifths the pitch of the stays.
C = 520 for plates fitted with stays, double nuts, and washers outside riveted to the plate. The washers to be the same thickness as the plate, and in diameter equal to two-thirds the pitch of the stays.
C = 568 for plates fitted with stays, double nuts, and doubling strips outside riveted to the plates. The strips to be the same thickness as the plate, and in width equal to two-thirds the pitch of the stays.
T = thickness of plate in sixteenths of an inch.
P = greatest pitch of stays.
p = least pitch of stays.
W = working pressure in lbs. per square inch.

Flat plates having doublings, at least two-thirds of their thickness, riveted to them.

$$\frac{C \times \left(T + \frac{t}{2} - 1\right)^2}{P^2 + p^2} = W.$$

Where t = thickness of doubling plates in sixteenths of an inch.
C, T, P, p, and W, as before.

Note.—For front plates in the steam space, which are not protected against the direct action of the flame, the constants given above are to be reduced 20 per cent.

Tube Plates—

$$\frac{320 \times (T - 1)^2}{P^2 + p^2} = W.$$

Where T and W are as before.
P = greatest pitch of stay tubes from centre to centre in inches.
p = least pitch of stay tubes from centre to centre in inches.

When girders are fitted to the tops of combustion chambers, the thickness of the tube plates must be found from the formula—

$$\frac{W \times L \times P}{(P - d) \times 1800} = T.$$

Where W = working pressure in lbs. per square inch.
L = width of combustion chamber over the plates in inches.
P = horizontal pitch of tubes in inches.
d = inside diameter of plain tube in inches.
T = thickness of tube plate in sixteenths of an inch.

Stays supporting Flat Surfaces—

$$\sqrt{\frac{S \times W}{C}} + \frac{1}{8} = D.$$

Where S = surface in square inches, supported by the stay.
W = working pressure in lbs. per square inch.
D = effective diameter of stay in inches.
C = 7700 for steel screwed stays.
C = 6500 for iron screwed stays.
C = 8900 for longitudinal steel stays.
C = 7000 for longitudinal iron stays.

Stay Tubes are not to be subjected to a greater stress than 7500 lbs. per square inch.

Circular Furnaces.—Thickness of plain furnaces and of furnaces with Adamson rings pitched more than 20 inches apart—

$$\frac{51\cdot5}{D}\{18\cdot75\,T - (L \times 1\cdot03)\} = W.$$

Thickness of bulb furnaces made by the Leeds Forge Company—

$$\frac{1250 \times (T - 2)}{D} = W.$$

Thickness of furnaces with Adamson rings pitched not more than 20 inches apart, or corrugated, ribbed, and suspension furnaces (Fox, Purves, Morison, Deighton, and Brown's cambered)—

$$\frac{1160 \times (T - 2)}{D} = W.$$

Thickness of Farnley spiral furnace and Holmes' furnace—

$$\frac{950 \times (T - 2)}{D} = W.$$

Where T = thickness of plate in sixteenths of an inch.
L = length of furnace or pitch of stiffening rings in inches.
D = smallest outside diameter of furnace in inches.
W = working pressure in lbs. per square inch.

Girders for Combustion Chamber Tops—

$$\frac{C \times d^2 \times T}{D \times L^2} \times \frac{(n + 1)^2}{n(n + 2)} = W, \text{ when } n = 2, 4, \text{ or } 6.$$

$$\frac{C \times d^2 \times T}{D \times L^2} \quad . \quad . \quad . \quad . \quad = W, \text{ when } n = 1, 3, \text{ or } 5.$$

Where C = 14,500 for steel and 13,180 for iron.
d = depth of girder at the centre in inches.
T = thickness of girder in inches.
D = distance from centre to centre of girders in inches.
L = length from tube plate to tube plate, or from tube plate to back of combustion chamber, in inches.
n = number of stays fitted with each girder.

GENERAL CONSTRUCTION.

1. Each boiler must have at least one glass water gauge, two test cocks, and one steam pressure gauge. Double-ended boilers are to have these fittings at each end. One salinometer cock must also be fitted to each boiler. Where the water gauge pillar is attached by pipes to the steam and water spaces, cocks or valves should be fitted to the shell of the boiler at the ends of these pipes.

2. A stop-valve is to be fitted to each boiler, so that any one of a series of boilers may be worked independently if required. The neck of the stop-valve to be as short as possible. All boiler and engine stop-valves to be tested to at least twice the working pressure.

3. Two safety-valves will be required for each main boiler, and they must be tested under steam to the satisfaction of the surveyors, and set to a pressure not more than 3 per cent. in excess of the intended working pressure. The combined area of the valves is to be sufficient to prevent the steam accumulating to more than 10 per cent. of the working pressure during fifteen minutes full firing, with main engines stopped. If the boilers be supplied with forced draught, the valve area must be increased so that the same conditions may be met.

4. Easing gear is to be provided, and so arranged that the safety-valves on any one boiler may be lifted simultaneously, without interfering with those on any other boiler.

5. Surface and bottom blow-off cocks should be fitted to the boiler, in addition to the cock on the hull of the vessel, and all cocks must have

spigots extending through the hull plating, with a plate flange round same on the outside.

6. It is recommended that manhole doors in boilers be not less than 16 inches by 12 inches.

7. Upon completion, the boilers and superheaters are to be tested by hydraulic pressure to twice the intended working pressure, and, after being placed in position in the vessel, they must be efficiently secured by brackets and stays to prevent any fore-and-aft or athwartship movement. It is strongly recommended, because of the rapid corrosion which takes place in material near them, that the boilers be kept as high as possible above the floors or tank top, and that the under side of the boilers be efficiently insulated.

8. Donkey boilers need not have more than one safety-valve, provided the valve area be not less than ½ square inch for each foot of grate surface. In other respects, the requirements for donkey boilers are the same as for main boilers.

APPENDIX G.

BUREAU VERITAS RULES FOR BOILER SHELLS, WORKING PRESSURE, OR THICKNESS OF PLATES, AND FOR SIZES OF STAYS.*

CIRCULAR SHELLS AND STEAM-HOLDERS WITH INTERNAL PRESSURE.

A RIVETED joint may fail through the tearing of the plate or butt strap between the rivets, the shearing of all the rivets, or by a combination of the two. The following formulæ apply to these several cases. The plate thickness and the diameter of rivets to be applied to have the highest values which each formula would give separately.

I. *Rupture through Plate.*

The formulæ for working pressure and plate thickness are in this case:—

$$\left.\begin{aligned} P &= \frac{2aR(t-0{\cdot}04)}{D} \\ \text{and} \qquad t &= \frac{PD}{2aR} + 0{\cdot}04 \text{ inch} \end{aligned}\right\} \quad - \quad - \quad - \qquad (I.)$$

Where P = allowed working pressure, above atmosphere, in lbs. per square inch.

D = greatest inside diameter of boiler shell or steam-holder in inches.

t = thickness of shell plates in inches. $t - 0{\cdot}04$ inch represents the thickness left after a reduction of 0·04 inch through corrosion.

R = the tensile stress in lbs. per square inch, which will be allowed in the plate. The value of R will be the breaking strength divided by 4, the latter figure representing the factor of safety for the plate after it has been corroded away by 0·04 inch.

If the actual breaking strength happens to be known by tests carried out to the Administration's satisfaction, it may be applied for finding R; but when, as usual, it is not known, the value of R will be:—

For Steel.—The fourth part of the lower limit of tensile strength chosen by the designer, which, in such case, is to be stated when a boiler design is submitted for approval.

For Iron.—11,200 lbs. per square inch, corresponding with a tensile strength of 20 tons. A table annexed shows the values of 2 R for various tensile strengths.

a = ratio of the resistance of the plate left between the holes to that of the full plates. It will be determined from the following expression:—

$$a = \frac{p-d}{p}.$$

---

* *N.B.*—If a boiler is intended for a vessel belonging to a country where the law prescribes heavier scantlings than those required by the following paragraphs, the builders have, of course, to comply with the legal requirements.

Where p = pitch of rivets in outer row in inches.

d = diameter of rivet holes in inches, either the real diameter or a corrected one, according to the following clauses :—

1. When the rivet holes are drilled, or when, having been punched, they are afterwards drilled or rimered out so that the injured metal around is completely removed, the real diameter may be taken.

2. When the holes are simply punched, they will be considered as being ¼ inch larger in diameter than as punched.

TABLE LXXI., showing the Values of 2 R.

In Formulæ (I.) *and* (IV.) *for Various Tensile Strengths of the Material.*

Tensile Strength of Plates in Tons per Sq. Inch.	Value of 2 R.	Tensile Strength of Plates in Tons per Sq. Inch.	Value of 2 R.
32	35,800	24½	27,400
31	34,700	24	26,900
30	33,600	23½	26,300
29	32,500	23	25,800
28	31,400	22½	25,200
27	30,200	22	24,600
26½	29,700	21½	24,100
26	29,100	21	23,500
25½	28,600	20½	22,900
25	28,000	20	22,400

II. *Rupture through Rivets.*

In this case the following are the formulæ for finding the allowed working pressure or required rivet section :—

$$\left.\begin{aligned} P &= \frac{2\,A\,s}{D\,l} \\ \text{and} \quad A &= \frac{P\,D\,l}{2\,s} \end{aligned}\right\} \quad - \quad - \quad - \quad - \quad \text{(II.)}$$

Where P and D have the same meaning as before, and

l = the length in inches of the identical parts into which a riveted joint can be subdivided. In most cases, l is the pitch of the rivets in the outer rows (figs. *g* and *h*). In general, it depends upon the system of joint adopted.

s = the maximum shearing stress in lbs. per square inch which will be allowed on the rivets. It will be the fourth part of the actual shearing resistance of the material, if known, from tests. If the actual shearing resistance of the rivet bars is not known, it will be assumed to amount to 0·8 of their tensile strength, and the value of s will be :—

For Steel.—The fifth part of the lower tensile limit adopted by the designer. The upper tensile limit may not in any case exceed 27 tons per square inch.

For Iron.—9000 lbs. per square inch, corresponding with a tensile strength of about 20 tons per square inch.

A = the total shearing surface in square inches of the rivets (that is, twice the area of the rivet hole when a rivet is in double shear), with or without corrections, according to the following rules:—

1. The total area of the rivet hole, without any reduction, may be brought into account when the rivet holes are drilled in place after the plates are bent, and the longitudinal and circumferential seams at least double-riveted, the former by machine.

This clause also applies to the case where the holes, having been punched, are afterwards drilled out in place, so as to correspond perfectly with each other.

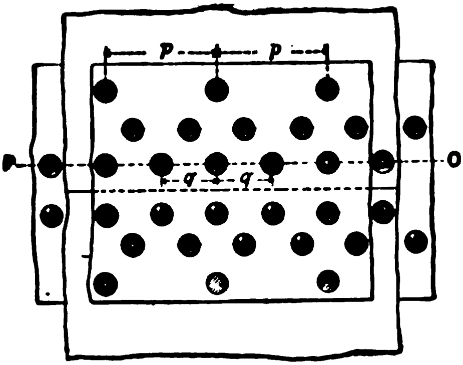

Fig. g.

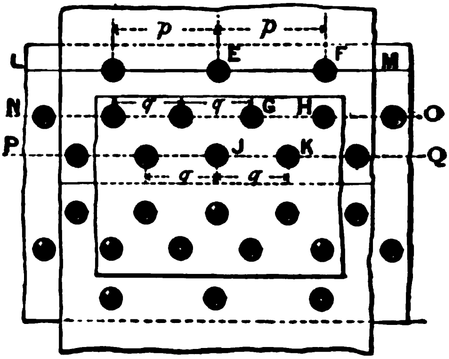

Fig. h.

2. $\frac{15}{16}$ of the full area is to be taken when the joint is made, as in the preceding case, with the exception that the riveting is done by hand.

3. $\frac{7}{8}$ of the full area are to be applied when the rivet holes are punched after the plates are bent, and the longitudinal seams at least double-riveted.

III. *Combined Rupture through Plate and Rivets.*

This case is only to be examined when the outer row has a wider pitch than the inner ones.

The formula to be applied in this case is:—

$$P = \frac{2(B \times R + C \times S)}{D \times l}. \quad - \quad - \quad - \quad \text{(III.)}$$

Where P, R, S, D, and l have the same meaning as before.

B = the sectional area in square inches of the plate on the portion, l, of the joint along the line of its supposed rupture, assuming that, in case the plate is liable to corrosion, its thickness has been reduced by 0·04 inch. The area of the plate is to be corrected as per Case I., No. 2, when the rivet holes are simply punched.

C = the total area of the rivets which are supposed to shear on the length, l, corrected, if required, in the same way as prescribed in Case II., Nos. 2 and 3.

For a rivet in double shear, the resistance will be considered as being twice that of one in single shear.

IV. *Rupture through Butt Straps.*

Rupture may take place along one of the inner rows of rivets (see P, Q, figs. *g* and *h*). The formulæ for this case, based on the same principle as (I.), are :—

$$\left.\begin{aligned} P &= \frac{2aR(t-0{\cdot}04)}{D} \\ \text{and} \qquad t &= \frac{P \times D}{2aR} + 0{\cdot}04 \text{ inch} \end{aligned}\right\} \quad - \quad - \quad - \quad (\text{IV.})$$

Where P, D, and R have the same meaning as before.

t = thickness in inches of butt strap, or sum of thicknesses, if there are two straps. (The thickness, of course, not to be less than required for caulking.)

$$a = \frac{q-d}{q}.$$

Where q = pitch of rivets in the inner row in inches.

d = diameter in inches of the rivet holes in the inner row; increased by $\frac{1}{4}$ of an inch, if the holes are simply punched.

V. *Combined Rupture through Butt Straps and Rivets.*

Formula (III.) applies to this case, B being the section of the butt strap, or straps, along which rupture would take place.

Corrections as for case III., Nos. 2 and 3.

Remarks.—No rivet holes to be nearer the edge of any plate than the diameter of the rivet. In zig-zag riveting, the distance between the rows is to be such that no rupture through plate or butt strap is to be feared along the zig-zag line.

When stays are bolted through the shell, they should be so arranged that they do not weaken the shell plates more than the riveted joints. If the resistance at the stay bolts is the smaller of the two, the plate's thickness shall be determined by it. It will be found from a formula the same as (I.), p and d applying to the stay bolts.

For circumferential seams, double riveting will be required if the thickness of the plates exceeds $\frac{1}{2}$ inch.

In double-ended boilers with six furnaces, treble riveting will be required for the circumferential seams connecting the courses with each other; it is not required for the end seams.

In double-ended boilers with four furnaces, the same arrangement is recommended.

Example showing the application of the above rules to the joint in fig. *h* (p. 654).

I. The value, $p-d$, is the clear distance between the rivet holes, E and F. If the holes are punched,

$$p - (\text{dia. hole E} + \tfrac{1}{4} \text{ inch}),$$

is to be considered as the equivalent length.

II. The length, l, is equal to p. The rivets, which would be sheared on this length, would be—

In single shear: one-half of rivet E, one-half of rivet F—total, one rivet E.

In double shear: one-half of rivets H and J, the whole of G and K—total three rivets: each counting double. Consequently,

A = area of one hole, E + 6 × area of one hole, G. A is to be multiplied by $\frac{15}{16}$ or $\frac{7}{8}$, in accordance with Case II., 2 and 3, if the conditions mentioned therein take place.

III. Rupture of plate along the line, N O, and shearing of the rivets in the outer row.

No corrosion is likely to take place in the interior of the joint. Consequently, in formula (III.)—

B = (p − $1\frac{1}{2}$ × diameter of rivet hole, G) × thickness of plate; the diameter of the rivet hole, G, increased by $\frac{1}{4}$ inch, if the holes are punched. The shearing surface of the rivets is one-half of E and one-half of F, therefore,

C = area of one rivet hole, E, to be multiplied by $\frac{15}{16}$ or $\frac{7}{8}$, if required, according to clauses Nos. 2 and 3 (Case II.)

IV. Rupture of butt straps through P Q.

The area which resists rupture is, on the length, q: (q − diameter of hole, J) × sum of thickness of straps, the diameter of J to be increased by $\frac{1}{4}$ inch, if the holes are simply punched.

V. Rupture of butt strap through L M.

In this case, the rivets in the rows, N O and P Q, must shear (double shear), and we have in formula (IV.)—

B = (p − diameter of hole, E) × thickness of wide strap. The diameter, E, being increased by $\frac{1}{4}$ inch, if the holes are simply punched.

C = 3 × area of hole, G + 3 × area of hole, K. Which value has to be multiplied by $\frac{7}{8}$ or $\frac{15}{16}$, if the Case II., Nos. 2 and 3, occur.

Shells of Superheaters.—The same mode of determining the working pressure or the thickness of shell plates will be followed for circular cylindrical superheaters, and the same formulæ may be used, but with the following alterations:—

1. When the plates are exposed to the direct action of the products of combustion, the values of R and S, as given for Cases I. and II., will have to be multiplied by 0·8 and the addition to the thickness of plates, on account of corrosion, will be increased from 0·04 to $\frac{3}{16}$ of an inch, to compensate for the corrosive action of the gases.

Formulæ (I.) become, therefore—

$$P = \frac{1{\cdot}6\,a\,R\left(t - \frac{3}{16}\right)}{D},$$

and

$$t = \frac{P\,D}{1{\cdot}6\,a\,R} + \tfrac{3}{16}\text{ inch.}$$

2. When the plates are protected from the direct action of the products of combustion, R and S will be multiplied by 0·9, and the additional thickness for burning away will be $\frac{1}{8}$ inch.

Formulæ (I.) become in this case—

$$P = \frac{1{\cdot}8\,a\,R\,(t - \frac{1}{8})}{D}, \text{ and } t = \frac{P\,D}{1{\cdot}8\,a\,R} + \tfrac{1}{8} \text{ inch.}$$

FLAT PLATES.

The allowed working pressure or the thickness of flat plates is to be determined by the following formulæ :—

$$P = \frac{(t-1)^2}{a^2+b^2} \times \frac{T}{C}, \text{ and } t = 1 + \sqrt{(a^2+b^2)\frac{P\,C}{T}}.$$

Where P = allowed working pressure above atmosphere in lbs. per sq. inch.
t = thickness of plate in sixteenths of an inch.
a = pitch of stays in inches, in one row.
b = distance in inches between two rows of stays.

Note.—When plates are effectively stiffened by doubling plates well riveted thereto, and having a thickness, t, in sixteenths, the value, $t + \frac{t}{2}$, may be substituted for t in the formula.

In case of irregular staying, such as in the annexed sketch (fig. *i*)—

$\frac{1}{2}(P_1 + P_2)$ shall be taken instead of $\sqrt{a^2 + b^2}$.

T = tensile strength of the material in tons per square inch of the original section.

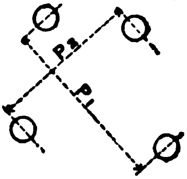

Fig. *i*.

It is to be determined in the same way as for shell plates, that is—

For steel it will be equal to the lower limit of the tensile stress, which is to be stated on the drawing.

For iron, 20 tons per square inch.

C = a constant, the value of which depends upon the mode of staying, as follows :—

C = 0·084 when the stays are screwed into the plates and riveted over.

C = 0·065 when the stays are screwed into the plates and fitted with outside nuts at both ends.

C = 0·062 when the stays are fitted with inside and outside nuts and washers, provided the diameter of the outside washers be at least 0·4 of the pitch between the rows of stays. The thickness of the washer to be at least ¾ that of the plate.

C = 0·055 when the stays are fitted with inside and outside nuts and washers, the outside washer being riveted to the plate and having ¾ of the plate's thickness and a diameter equal to 0·6 of the pitch between the rows of stays.

C = 0·050 when the outside washers are replaced by strips of plate, having a width of at least 0·6 of the distance between the rows of stays, with a thickness not less than ¾ that of the plate ; the strip being well riveted to the plate.

For the values of $\frac{T}{C}$, see Table lxxii.

When the plates are in contact with steam on one side and flame or hot gases on the other, the thickness is to be increased.

For instance, when, in return tube boilers, the top front plates are in no way protected from the hot gases, the working pressure or thickness will, in such a case, be determined by the formulæ :—

$$P = \frac{(t-2)^2}{a^2+b^2} \times \frac{0{\cdot}9\,T}{C},$$

and

$$t = 2 + \sqrt{(a^2+b^2)\frac{P\,C}{0{\cdot}9\,T}}.$$

When the said front plates are protected by a flame plate, no increase of thickness will be required.

When front plates are in two pieces, the lap should be double-riveted if the thicker plate is $\frac{1}{2}$ inch or above.

TABLE LXXII.—VALUES OF $\frac{T}{C}$ IN THE FORMULÆ FOR FLAT PLATES.

Tensile Strength of Plate in Tons per Sq. Inch.	C = 0·084.	C = 0·065.	C = 0·062.	C = 0·055.	C = 0·050.
20	$\frac{T}{C}$ = 238·0	$\frac{T}{C}$ = 307·6	$\frac{T}{C}$ = 322·6	$\frac{T}{C}$ = 363·6	$\frac{T}{C}$ = 400
21	250·0	323·0	338·6	381·8	420
22	262·0	338·4	354·8	400·0	440
23	273·8	353·8	371·0	418·0	460
24	285·8	369·2	387·0	436·2	480
25	297·6	384·6	403·2	454·4	500
26	309·6	400·0	419·4	472·6	520
27	321·4	415·3	435·4	490·8	540
28	335·4	430·7	451·6	509·0	560
29	345·2	446·1	467·8	527·2	580
30	357·2	461·5	483·8	545·4	600

STAYS.

The diameter of stays supporting flat surfaces is to be determined by the following formula :—

$$d = \tfrac{1}{8}\text{ inch} + \sqrt{\frac{Q}{300\,T}}.$$

Where d = effective diameter in inches (for instance, the diameter at bottom of thread in screw stays).

Q = total load on stay in lbs.

T = tensile strength of the material in tons per square inch.

For steel, this tensile strength will be the lower limit chosen by the boiler designer; for iron, it will be taken at 22 tons. In both cases the actual strength may be applied, if it is known from tests.

If the stays are not round, their cross-section must be such that the stress per square inch, caused by the load, Q, nowhere exceeds 5·75th part of the tensile strength, after deducting $\frac{1}{16}$ of an inch all round as an allowance for corrosion or wear.

In welded stays, the stress, as just described, will be reduced by 20 per cent. Welding of steel stays is only allowed for very mild qualities.

For high working pressures, such as used in triple-expansion engines, it is recommended to screw all stays into the plates they support, in addition to fitting them with nuts.

This also applies to stay tubes, with the exception that it is recommended that nuts should not be fitted in combustion chambers.

CIRCULAR FURNACES.

Plain Cylindrical Furnaces.

The working pressure and the thickness of the plates may be calculated from the following formulæ :—

$$P = \frac{C t^2}{D L},$$

and

$$t = \sqrt{\frac{D P L}{C}}.$$

Where t = the required thickness of plates in inches.
D = outside diameter of furnace in inches.
P = working pressure in lbs. per square inch (above atmosphere).
L = length of furnace in feet; or, if made or fitted with efficient rings, the length between the rings.
C = 70,000 when the furnace is truly circular and the longitudinal seams are welded, butted, or lapped and bevelled, and double-riveted in the last-named case.
C = 60,000 when the furnace is not truly circular, or when the longitudinal seams are simply lapped.

The above constants apply to iron plates of fair quality.

They may be multiplied by 1·2 when the plates are of mild steel, or of iron of the best quality with regard to behaviour under the action of the fire, having a uniform structure without blisters or other defects, and fulfilling the following conditions, to be ascertained by testing samples :—

	With the Grain.	Across the Grain.
Tensile strength, - - - - - - -	23 tons per sq. in.	21½ tons per sq. in.
Elongation in 8 inches, - - - - - -	16 per cent.	10 per cent.
Cold bending without fracture to an angle of	60°	35°
Hot ,, ,, ,,	180°	180°

The thickness should in no case be less than—

$$t = \frac{P D}{8000}, \text{ for iron plates,}$$

$$t = \frac{P D}{9000}, \text{ for steel plates.}$$

It is recommended that plain furnaces should be divided into rings of such a length that the thickness of plates may not exceed ⅝ inch.

DOMED FURNACES IN VERTICAL BOILERS.

When the tops of furnaces in vertical boilers are portions of spheres, the thickness must not be less than:—

$$t = \frac{Pr}{8000} + \cdot 15 \text{ for iron plates, or } t = \frac{Pr}{9000} + \cdot 15 \text{ for steel plates.}$$

CORRUGATED AND RIBBED FURNACES.

The plate thickness is to be found by the following formulæ:—

1. For corrugated furnaces—

$$T = \frac{PD}{1259} + 2.$$

Where P = working pressure in lbs. per square inch.
T = thickness of plate in sixteenths of an inch.
D = outside diameter in inches measured on the top of the corrugations.

The formula applies to corrugations 6 inches long and 1½ inch deep.

2. For ribbed furnaces, when manufactured to the satisfaction of the Administration—

$$T = \frac{PD}{1160} + 2.$$

Where P and T are as above.
D = outside diameter of plain parts between the ribs in inches. The formula applies to ribs spaced 9 inches and projecting 1⅜ inch, the difference between the greatest and the smallest diameters in any part of the furnace not exceeding $\frac{3}{1000}$.

3. For bulb furnaces—

$$T = \frac{PD}{1259} + 2.$$

Where D is the outside diameter in inches measured between the bulbs.

The coefficients in the above formula apply to the case where the tensile strength of the material is 26 to 30 tons per square inch. When it is below 26 tons the coefficient is to be reduced $\frac{1}{26}$ for each ton below 26.

Combustion Chamber Girders.

The strength of girders on the tops of combustion chambers shall be determined as follows:—

$$P = \frac{C d^2 t}{(W - p) D L}.$$

Where P = working pressure.
C = a constant found as under.
N = number of bolts in each girder.
d = depth of girder in inches.
t = thickness of girder in inches.
p = pitch of bolts in girder in inches.
L = length of girder between supports in inches
D = distance between girders, centre to centre, in inches.
W = width of firebox, from tube plate to back plate, in inches.

1. For iron girders—

$$C = \frac{12{,}000\ N}{N + 1} \text{ for odd numbers of bolts.}$$

$$C = \frac{12{,}000\ (N + 1)}{N + 2} \text{ for even numbers of bolts.}$$

2. For steel girders—

C, as formed above, may be increased 10 per cent.

GENERAL AS TO BOILER ARRANGEMENTS AND FITTINGS.

When two or more boilers are fitted, they should be arranged to work separately or independently of each other, either by stop-valves between the boilers and the common superheater, or by stop-valves between the separate superheaters and the main steam pipes.

Each boiler must be fitted with a separate pressure gauge. Double-ended boilers to have a pressure gauge at each end.

Steam pipes of donkey engines must be independent of the main pipes, so as to keep the steam off the main engines when only winches or other auxiliary engines are working.

The bottom blow-off to be arranged with one cock directly attached to the shell of the boiler, and another directly attached to the skin of the vessel. The surface blow-off must be similarly arranged. The main stop-valve to be so situated and fitted that it can be worked from the starting platform or the stokehole floor.

To protect the plating, the cocks on the ship's bottom should be fitted with spigots passing through the plating and through a flange on the outside. If this flange is of iron it must be galvanised.

Steam domes or superheaters, when placed in the uptake and exposed to the direct impact of the flame, will only be allowed as an exception, and must be efficiently protected by flame plates. In all cases it must be possible to examine efficiently the interior and the exterior of the domes or superheaters.

To prevent the boilers shifting in a transverse direction through the rolling of the vessel, or longitudinally in case of collision, they must be properly secured in their seats.

All manholes to be fitted with compensating rings.

At least two safety-valves of an approved design must be fitted to each main boiler.

Their total area is given by the following formula :—

$$A = \frac{8{\cdot}7}{\sqrt[4]{(P - 18)^3}}.$$

Where A = sectional area of safety-valve in square inches per square foot of grate surface.

P = working pressure of boiler in lbs. per square inch.

When forced draught is provided for, the area is to be increased in proportion to the increased evaporative power of the boilers.

Suitable arrangements and gear to be fitted in connection with the safety-valves, whereby they may be lifted from the deck as well as from the stoke-hole floor.

If it be practicable to isolate a superheater communicating simultaneously with two or more boilers, it will be necessary to furnish it with a safety-valve of suitable dimensions.

APPENDIX H.

ADMIRALTY RULES FOR STEEL BOILERS AND MATERIALS AND THEIR TESTS.

ADMIRALTY rule for working pressure provides that the water-pressure test shall not exceed $\frac{4}{9}$ of the ultimate strength of the shell, or a factor of safety of 2¼, when subjected to water test pressure, and the working steam pressure is fixed at 90 lbs. below the test pressure, which is called their "constant margin" of safety for all pressures.

SUPERVISION OF BOILER WORK.

The following instructions to boiler-maker overseers are those usually given in Admiralty specification:—

The boilers will be subject to the supervision of an overseer, who will be directed to attend on the premises of the contractors during the progress of the work on the boilers, to examine the material and workmanship used in their construction, to witness the prescribed tests, and to see that this specification, as regards the boilers and work in connection, is conformed to in all respects by the contractors. The extent of supervision is described in the following paragraphs extracted from Admiralty instructions to overseers, and the contractors are to afford him every facility for their proper execution.

The plates and other material used in the construction of the boilers to be subjected to such tests as may be directed in the specification. Every plate used is to be carefully examined by the overseer for laminations, blisters, veins, and other defects, and to ensure that it is of the proper thickness and brand. No plate, angle, &c., which from any cause is considered by the overseer to be unfit for the intended use is to be fitted.

During the construction of the various parts of the boilers, the overseer is to satisfy himself that the dimensions as shown on the approved drawings are being adhered to by the contractors.

Whenever plates are flanged or welded, or in any case where iron or steel is worked in such manner that it is particularly liable to suffer in strength unless carefully handled, the overseer is to be present if possible on all occasions during the time the work on each article is in progress, and he is to fully satisfy himself that it is sound before he allows any part to be put in the boilers.

Samples of the rivets being used for the boilers are to be taken by the overseer during the progress of the work and tested as specified hereafter, and any batches of rivets found defective are to be rejected. Before rivets are put in, the overseer is to see that the plates are brought properly together, and that the holes are fair with one another. He is not to allow drifting on any account, but he is to see that they are carefully rimmed fair where necessary. He is also to make sure during the progress of the work that the rivets fill the holes completely, and that the heads are properly set up, well formed, and finished.

The overseer is to see that all internal parts of the boiler are riveted with rivets having heads and points of approved shape, and that any seams he considers necessary are riveted on the fire side. No snap heads are to be allowed in the internal parts. Any proposal for hydraulic riveting the internal parts is to be submitted to the Admiralty, with sketch of the proposed heads and points. In all parts where the rivets are not closed by hydraulic riveting machinery he is to see that the rivet holes are countersunk and that coned rivets are used. All holes in the plates, angles, &c., are to be drilled, and not punched, and are to be drilled in place after bending. The clearance between rivet hole and rivet before closing is not to be greater than approved by the overseer.

The overseer will see that the particulars of the form, dimensions, and pitch of the various stays shown on the drawings are adhered to, and samples of them are to be tested as directed in this specification; and he will be guided by his experience as a workman in testing and judging of the soundness of the forging and construction of the various stays.

He is to see that palm stays, if fitted, are forged from the solid and not welded, that all short stays are nutted on all flat surfaces except where otherwise approved and screwed to a pitch of twelve threads per inch for stays of 1 inch diameter and above, that the holes for the screwed stays in the water spaces are drilled and tapped together after the furnaces and combustion chambers have been riveted in place in the boiler, that the combustion chamber stays are drilled square to the bevel of the combustion chamber plates, and that no bevel washers are inside the chamber. Any girder stays used for combustion chambers are to be well bedded on to the tube plates to the satisfaction of the overseer.

The overseer is to see that the arrangement of the zinc plates shown on the approved drawings is adhered to, that the metallic surfaces in contact are filed bright, and that means are adopted to secure a firm grip of the clips by which the plates are attached.

The overseer is to witness the testing, in all cases, of the boiler tubes, in accordance with this specification, before they are put in the boiler.

When the boilers are reported to the overseer by the contractors as being completed, ready for testing by water pressure, the overseer is to witness a preliminary test of them in accordance with the specification, carefully observing with the assistance of gauges, and straight edges whether any bulging or deflection of the plates has taken place.

The official test will be conducted on all occasions in the presence of an inspecting officer. A test pressure gauge is supplied to the overseer from the Admiralty, and the official test is to be made with this gauge.

After the boilers have been tested by water pressure the overseer is to see that they are properly cleaned inside and outside, and then well painted with red lead. It is important that the whole surface of the boilers should be thoroughly cleansed of scale formed in manufacture before any paint is put on them. The boilers are not to be exposed to the weather till they are so painted, and properly cleaned and closed up to his satisfaction.

The overseer is to make himself fully acquainted with the progress of the whole of the work in its various stages, to satisfy himself that every part is sound before it is allowed to be put in the boilers, and to see that the following instructions for the treatment of mild steel are strictly complied with.

QUALITY OF AND TESTS FOR MATERIALS.

(a) *Steel generally.*

Quality of. All steel used throughout the whole of the work is to be of quality to be approved by the Admiralty. The names of the firms from whom it is proposed to obtain this material are to be submitted for approval before the order is placed. The quality and makers' names of all steel are to be marked on all drawings sent to the Admiralty.

Solid ingots for forgings. All steel forgings are to be made from solid ingots, and test pieces from each important completed forging will be subject to the tests for steel enumerated below, and facilities are to be afforded, by a proper system of numbering and registering the ingots, to enable the origin of any forging to be ascertained.

Testing in presence of Admiralty officers, &c. The whole of the temper and bending tests for furnace and other plates exposed to flame, also those for the important castings and forgings, and not less than 10 per cent. of all the other tests enumerated, are to be made in the presence and to the satisfaction of the Admiralty officers. These tests are intended to facilitate supply, but are not to preclude examination at the machinery contractors' works, and rejection of any plates or castings found defective after delivery. The tensile strength and elongation per cent. are to be stamped on each plate.

Copies of invoices to be supplied, &c. Copies of the invoices upon which the plates, bars, angles, rivets, stays, tubes, &c., for the boilers have been received, with the names of the manufacturers of the material and their tests, are to be supplied to the boiler overseer for his use in accepting them. When any orders are placed with steel makers, copies of the orders showing the particular purpose of the materials should be sent to the Admiralty overseer, with a copy of the specified tests.

(b) *Plate, Angle, and Bar Steel, for Boilers and any Steel Steam Pipes.*

Tests for steel plates, &c. Tensile test. Every plate, &c., used is to be tested as follows:—Strips cut lengthwise or crosswise are to have an ultimate tensile strength per square inch of section as follows:—(1) Plates, &c., not exposed to flame, not less than 27 tons, and not exceeding 30 tons; (2) lower front plates not less than 25 tons, and not exceeding 28 tons; (3) furnace plates not exceeding 25 tons, and not less than 23 tons; (4) fire-box and other plates exposed to flame not exceeding 26 tons, and not less than 24 tons. The elongation taken on a length of 8 inches, must be at least 20 per cent. for strips from shell and other plates which are not exposed to flame and which will not be flanged, and at least 25 per cent. for plates which are not exposed to flame and which will be flanged, at least 27 per cent. for strips from the furnaces, and at least 26 per cent. for strips from fire-box and other plates exposed to flame. Edge shearings, to bring plates to proper dimensions, are to be equal on opposite sides.

Temper tests. Strips cut lengthwise or crosswise, 1½ inches wide, heated uniformly to a low cherry red, and cooled in water of 82° Fahrenheit, must stand bending double in a press to a curve of which the inner radius is one and a-half times the thickness of the steel tested.

Welding and forging tests of plates exposed to flame.

The plates for furnaces and other parts exposed to flame are, in addition, to be tested by welding and forging, and some of the welded pieces are to be broken in the testing machine to ascertain the efficiency of the welding.

Test pieces.

The pieces cut out for testing are all to be cut in a planing machine, to have the sharp edges removed, and are to be of parallel width for at least 8 inches of length.

Forged tests of angle, &c., steel.

The angle, tee, and bar steel are to stand such other forge tests, both hot and cold, as may be sufficient, in the opinion of the overseer, to prove the soundness of the material and fitness for the service intended.

Hammering test.

The furnaces are to be subjected to a hammering test if required before the machinery is accepted. The test is to be carried out in accordance with Admiralty practice.

(c) *Steel Rivets for Boilers.*

Manufacture.

The whole of the rivets are to be properly heated in making, and care is to be taken that the finished rivets cool gradually.

Tensile test

The rivets are to be made from steel bars, the ultimate tensile strength of which should not be less than 25 tons, and should not exceed 27 tons per square inch, with a minimum elongation of 25 per cent. in a length of 8 inches.

Temper tests.

Pieces cut from the bars, heated uniformly to a low cherry red, and cooled in water of 82° Fahrenheit, must stand bending double in a press to a curve of which the inner diameter is equal to the diameter of the bar tested.

Forge tests.

Samples from each batch of rivets are to be selected and are to stand the following forge tests satisfactorily without fracture :—The rivet to be bent double, cold, to a curve of which the inner diameter is equal to the diameter of the rivet tested ; the rivet to be bent double when hot, and hammered till the two parts of the shank touch ; the head to be flattened when hot, without cracking at the edges, till its diameter is two and a-half times the diameter of the shank. Finally, the shank of a rivet is to be nicked on one side, and bent over to show the quality of the metal.

(d) *Boiler Tubes.*

Boiler tubes.

The tubes are to be made from acid or basic "open hearth" steel. Strips cut from the tubes must have a tensile strength not exceeding 25 tons and not less than 21 tons per square inch, with an extension of at least 25 per cent. in a length of 8 inches. Strips cut from the tubes flattened, heated to a blood heat and plunged into water 82° Fahrenheit temperature, should be capable of being doubled over a radius of $\frac{1}{2}$ inch without fracture. Pieces 2 inches long, cut from the ends of tubes under $\frac{8}{16}$ inch thick are to be capable, when cold, of being hammered down endwise until their length is reduced to $1\frac{1}{4}$ inch, and of being flattened until the sides are close together in each case without fracture. The ends of the tubes under $\frac{3}{16}$ inch thick are to admit of being expanded cold by a roller expander, worked in a tube hole to an increase of diameter of 15 per cent., and hot by a solid drift to an increase of diameter of 20 per cent. Tubes $\frac{3}{16}$ inch and over in thickness are to admit of being expanded hot and cold to half the increases of diameter required for tubes under $\frac{3}{16}$ inch thick. The above tests are to be applied to 2 per cent. of the tubes, to be selected by the examining officers. The failure of the tubes selected to stand the specified tests in a satisfactory manner may render the whole of any delivery liable to rejection.

APPENDIX I.

GERMAN GOVERNMENT RULES FOR BOILERS.

REGULATIONS RESPECTING STEAM BOILERS.

1. *Plating of Boilers.*—When the smallest dimensions of a cylindrical boiler exceeds 25 centimetres, or of a spherical-shaped boiler 30 centimetres, the portions of the boiler exposed to the flame, furnaces, and tubes shall *not* be made of cast iron.

Brass (copper) plates are only to be used for furnaces where their smallest dimension does not exceed 10 centimetres.

2. *Furnaces, Flues, and Combustion Chambers.*—Flues running round or through steam boilers must, at their highest point, be at least 10 centimetres below the minimum water level. This minimum level must, in lake and river vessels, be maintained when the vessel is inclined at an angle of 4°, and for sea-going ships at an angle of 8°. In boilers with a breadth of 1 to 2 metres, the distance to the water level must be at least 15 centimetres, and in boilers of greater breadth at least 25 centimetres.

These conditions do not apply to boilers in which the tubes are less than 10 centimetres diameter, nor to those wherein the portions of the plates in contact with steam are not liable to become red hot. The risk of plates becoming red hot may be assumed to be done away with if the heating surface exposed to the flame, before the flame reaches the point in question, is, with natural draught 20 times and with forced draught 40 times, greater than the area of the grate. The highest point in marine boiler flues, &c., is the *upper* surface of the plates in question.

3. *Feed Apparatus.*—Each boiler must have a feed valve which will close by boiler pressure when the feed is off. Where only one feed check-valve is provided shut-off valves must be fitted in the feed delivery pipes.

4. Each boiler must be fitted with two independent feed apparatus, each worked separately, and each to be capable, by itself, of keeping the boiler supplied. Several boilers connected together and used for the same purpose may, in this respect, be regarded as one boiler. Each feed apparatus to be capable of delivering 30 litres of water per square metre of heating surface per hour. Apparatus for boilers worked under forced draught, or having heating surface less than 35 times the grate area, must be proportionately larger. Each feed apparatus must have an independent suction pipe, or else a suitable arrangement of switch shut-off valves must be fitted. An auxiliary feed apparatus for a main boiler may be used for feeding a donkey boiler if there are independent delivery pipes, or if suitable switch shut-off valves are provided. If an injector is used for auxiliary feed purposes it must be capable of working satisfactorily with both high and low pressure steam; if it will not so work, some other provision must be made for use when pressure is low. Where a main boiler is used for driving winches in port, the main engine feed pumps will not be reckoned as a secondary feed apparatus. If supplementary feed arrangements, beyond those required by

the regulations, are provided, the piping for them must either be independent of that belonging to the regulation apparatus, or must be capable of being shut off from the latter by valves. Each main feed-pump must be fitted with an escape valve that cannot be shut off from the pump.

5. *Water Gauges.*—Each boiler must be fitted with a gauge glass, and also with a second means of ascertaining the height of the water. Each of these fittings must have a separate connection to the boiler, or if they have a common connection, the stand pipe must be at least 60 square centimetres in area, or 88 millimetres in diameter. Pipes for connecting water gauges to boilers must not be less than 16 centimetres in area, or, say, 45 millimetres in diameter. Long or much bent pipes must be larger in proportion. Internal connecting pipes must not be used, but two pipes may be led to one opening in the plating, provided the area of opening be made equal to the combined areas of the pipes. Gauge glasses are to be placed so as to be easily visible. The mark indicating lowest water level should not be higher than half-glass, and the mark showing highest point of heating surface should be above upper edge of lower nut which secures glass. The cocks must be easy to open and shut, and it must be possible to replace a broken glass without risk to the operator.

6. Water test cocks to be fitted: the lowest one must be placed at the level of the regulation water-line. All gauge cocks must be so fitted as to have a straight way through for cleaning them from scale or salt. Spindle valves must not be used.

7. *Water Level.*—The regulation lowest water level is to be distinctly marked on the gauge glass, and also in a prominent place on the boiler shell. The level of the flue, at its highest point, is also to be plainly and permanently marked on the shell of the boiler in way of the ship's beam.

Two water-gauge glasses are to be fitted on the boiler at the standard level of the water when the vessel is inclined as above, one on each side of the boiler, as far apart as possible, and symmetrically placed right and left of centre line of boiler.

When these two gauge glasses are fitted, the additional means named in Rule 5, for ascertaining the water level, may be dispensed with.

The marks indicating lowest water level are to be not less than 5 centimetres long, and are to be cut into the plating near to the water gauges. They are also to be distinguished by having the letters N. W. stamped on them. They must be on before the inspection of the boiler is made, and must be visible after it is lagged. Where water gauges of main boilers of river steamers and barges and of donkey boilers are direct on the boiler shell, a plate must be fitted at the lowest water-level mark with "Niedrigster Wasserstand" on it, and an arrow pointing to the glass. Where water gauges are connected to boilers by pipes (as must be the arrangements in all sea-going steamships) a similarly marked plate is to be fixed to the stand-pipe. The highest point of the heating surface is to be indicated as follows:—Where the water guage is fitted direct to the boiler shell a plate indicating the point in question, and marked "Hochster Feuer berührter Punkt," is to be fixed near the gauge. Where the gauge is connected to the boiler by pipes a similar plate is to be affixed to the gauge standard. These plates are to be permanently secured—not with removable screws.

In cases where the fore and aft trim of the vessel is subject to considerable variation, the minimum water level (paragraph 2) must either be raised

10 centimetres, or the mark indicating highest point of heating surface must be raised a similar amount. The permanent mark showing regulation lowest water level must, however, correspond with the particulars given on the maker's name-plate, as hereafter specified.

A third water gauge must be fitted on every double-ended boiler.

8. *Safety-Valves.*—Each boiler must be fitted with, at the least, one reliable safety-valve.

When several boilers have a common steam chest, from which they cannot be separately disconnected, two safety-valves will be sufficient.

Boilers of steamships, locomotives, and portable engines must have at least two safety-valves.

In steamships, except those that are sea-going, one valve is to be placed in such a position that the load on it can easily be ascertained from the deck.

The valves must be so arranged that they can be readily lifted at any time. They are to be loaded so that they will blow off immediately the working pressure is reached.

Safety-valve areas must be such that after firing up for 15 minutes the pressure does not accumulate more than 10 per cent.

Provided that the springs are satisfactorily fitted and loaded, the area of safety-valves, in square millimetres, may be as follows for each square metre of heating surface:—

Pressure above atmosphere,	5	6	7	8	9	10	11	12	13	14	15	16
Area in square millimetres,	131	112	98	86	79	72	66	60	56	54	52	51

For boilers worked under forced draught, or in which the heating surface is less than 35 times the grate surface, the safety-valve areas must be correspondingly increased. Safety-valve boxes must be fitted with drain pipes that cannot be shut off. The waste steam pipes from all safety-valves must blow off into the open air. Blowing off into a tank, or into the chimney, will not be permitted. The thickness of the washers which determine the compressions of the safety-valve springs will be fixed by the Official Surveyor and recorded in his book, and no alterations must be made except by his authority. Exception is made, however (by the regulations of 19th December, 1883), in the case of long voyages, and the chief engineer of the vessel is permitted to re-adjust the washers in case the valves blow off too freely. To meet this case the washers may be fitted in halves.

9. *Pressure Gauge.*—Each boiler must have a reliable pressure gauge on which the highest regulation working pressure is to be plainly marked.

Boilers of steamships must have two pressure gauges—one clearly visible to the fireman, the other, except when the vessel is sea-going, in a convenient position for being observed from the deck. If a vessel has several boilers connected to a common steam chest, one pressure gauge on deck will be sufficient, in addition to the one on each boiler.

Gauges to be placed where they can easily be seen. Dials to be marked in kilogrammes per square centimetre (or in atmospheres,—1 atmosphere equals 1 kilogramme per square centimetre), and marking must go up to 50 per cent. above the regulation working pressure. In vessels having more than one boiler, one pressure gauge (in prominent position in the engine room), in addition to those on the boilers, will suffice. The pipe to this gauge must be connected to all boilers, but must be capable of being shut off from each.

10. *Name Plate on Boiler.*—(1) Each boiler must have the highest working pressure, the maker's name, the shop number of the boiler, and the date of completion, plainly and permanently stamped on it. Steamships, moreover, must also have the standard lowest water level marked on.

(2) Boilers that have already been constructed, at the date of these regulations, are not required to be altered to comply with them.

(3) The regulations for boilers of steamships apply in all cases in which boilers are permanently fixed in, or connected with, the vessel.

The address of the maker must be given as well as his name.

10*a*. These particulars must be stamped on a metal plate, which is to be fixed on the boiler by copper rivets, the heads of which are to be not less than 12 millimetres in diameter. The plate to be in such a position that it is easily visible when there are casings on the boiler.

11. *Testing of Boilers.*—Every new boiler must, on completion, be tested by water pressure.

The test pressure for boilers intended for a working pressure of not more than 5 atmospheres to be double the working pressure, and for boilers intended for a working pressure of more than 5 atmospheres, the test pressure is to be 5 atmospheres above the working pressure.

Fig. *j*.

For boilers where the working pressure is under atmospheric pressure, the test pressure is to be 1 kilogramme per square centimetre. The plates of the boiler must withstand the test pressure without showing any permanent change of form, and must also remain tight under the pressure.

The boiler is to be regarded as leaky, if the water at the highest pressure leaks through in other forms than dewy moisture or pearly drops.

11*a*. When the boilers have been tested and found satisfactory, the copper rivets which fix the brass plate are to be officially stamped, and an impression of this mark is to be inserted on the certificate of test.

12. When boilers have undergone a thorough repair in the boiler shop, or when they have had to be stripped and extensively repaired on board, they must be tested by hydraulic pressure the same as new boilers.

When in boilers with internal furnace, the furnace has been taken out to repair or renew, and when in locomotives the fire-box has been taken out to repair or renew, or when in cylindrical boilers one or more plates have been renewed, the boilers must, on completion of such repairs, be tested by hydraulic pressure as above.

When a boiler has undergone considerable repairs the steam pressure to be allowed must be determined afresh.

13. *Test Gauge.*—The test pressure is to be registered by an open mercurial gauge, or by the standard pressure gauge provided by the inspecting official.

Each boiler to be tested must be fitted with a connection to suit the

standard gauge, and by which the gauge may be applied by the inspecting officer.

To facilitate application of official test gauge, a flanged branch is to be provided immediately below the usual pressure gauge. The opening in the flange and cock must not be less than 7 millimetres, the length of the branch must be 60 millimetres, and its breadth 25 millimetres. Form to be as shown in sketch.

RULES FOR STRENGTH OF BOILERS.

5th August, 1890.

The quantities are expressed in centimetres and kilogrammes:—

(1 kilo. per sq. cm. = 1 atm. = 14·223 lbs. per sq. in English).

Formulæ for thickness of shell plate and working pressure:—

$$\text{Working pressure} = \frac{S \times R \times 2T}{D \times F}.$$

$$\text{Thickness of plate} = \frac{P \times D \times F}{S \times R \times 2}.$$

Where P = highest steam pressure (kilos. per sq. cm.).
S = lowest tensile strength of material.
T = thickness of shell plate.
D = greatest internal diameter of boiler.

$$F = \begin{cases} 4{\cdot}75 \text{ when rivet holes in longitudinal seams are drilled and rivets closed by hydraulic pressure.} \\ 4{\cdot}5 \text{ when longitudinal seams are double strapped and have holes drilled and rivets closed by hydraulic pressure.} \end{cases}$$

$$R = \frac{\text{Pitch} - \text{Diameter of rivet}}{\text{Pitch}}.$$

$$\text{Or,} \quad R = \frac{\text{Area of rivet} \times \text{No. of rivets in pitch} \times a}{\text{pitch} \times \text{thickness of plate}} \times 0{\cdot}9 \times \frac{S_1}{S}.$$

$$\text{Where } a = \begin{cases} 1 \text{ for lap and single riveted butt joints.} \\ 2 \text{ for double riveted, \&c., butt joints.} \end{cases}$$

S_1 = Tensile strength of rivet material. Unless actual test has been made, this is to be taken as 3300 for iron, and 3800 for steel; but it must not exceed 4800 kilos. per sq. cm. in any case.

The lower of these two values of R must be used.

S is to be taken as 3300 for iron; for steel the lowest value given by the tests is to be used, but this must not exceed 5000 kilos. per sq. cm., and the material must show an elongation of at least 20 per cent. in a length of 20 centimetres. Steel of higher tensile strength may only be used under specially arranged conditions as to tests, &c.

In riveted joints the shearing strength of the rivets must not be less than the tensile strength of the plate left between holes.

Butt-straps are to be of the same quality as the shell plates, and of not less than 75 per cent. of their thickness.

In boilers without a central circumferential seam the outside butt-strap must not be thinner than the shell plate. If the shell plates are thicker than 1·25 centimetres, the central circumferential seam must be double riveted, and if they are 2·5 centimetres or more it must be treble riveted.

Rivet holes must be at least one diameter of rivet from edge of plate. Diameter of rivet hole must not be less than thickness of plate, or more than 2 × thickness of plate, the former limit to apply to thick plates and the latter to thin ones.

Diagonal pitch must not be less than 2·1 × diameter of rivet.

Section 2.—*Cylindrical Furnaces.*

The thickness of plate for cylindrical furnaces is given by the formula—

$$\text{Thickness} = K\sqrt{P \times L \times D} + 0{\cdot}2 \text{ cm.}$$

Where P = working pressure (kilos. per sq. cm.).
L = length between stiffeners (centimetres).
D = external diameter of furnace (centimetres).
K = 0·0033 when the longitudinal joints are welded and fitted with double butt-straps, and the furnace is truly cylindrical.
K = 0·0035 when the longitudinal joints are lapped only.

The thickness of plate must not, in any case, be less than that given by the formula—

$$\text{Thickness} = \frac{P \times D}{C} + 0{\cdot}2 \text{ cm.}$$

Wherein C = 670 for plain furnaces without Adamson rings or other stiffeners.
C = 820 for furnaces with one Adamson ring, when the unsupported distance between the ring and the ends is not more than 122 cm.
C = 920 for furnaces with two Adamson rings, when the unsupported distance between the stiffeners is not more than 79 cm.
C = 1160 for Fox or Morison corrugated, or Purves ribbed furnaces. In corrugated furnaces D is the external diameter of the depressed corrugation, and in ribbed furnaces the external diameter of the plain portion between the ribs is to be taken.

The thickness proposed for any type of furnace, such as Farnley, Deighton, &c., must be referred for special consideration.

For the boilers of river steamers the constant addition may be 0·1 cm. in place of 0·2 cm.

Section 3.—*Screw and Through Stays.*

The strength of screw and through stays must not be taken higher than—

500 kilos. per sq. cm. for iron.
600 „ „ „ steel.

Steel stays must not be welded.

In welded iron stays the section must be increased 20 per cent.

The pitch of screw stays in combustion chamber backs must not exceed 20 cm., and that of through stays in the steam space 45 cm.

Section 4.—Flat Surfaces.

The thicknesses of stayed flat surfaces are to be those given by the formula—

$$\text{Thickness} = 0{\cdot}15\ \text{cm.} + p\sqrt{\frac{P}{K}}.$$

Wherein p = greatest pitch of stays (in centimetres).
P = working pressure (kilos. per sq. cm.).
K = 2200 for stays screwed in and riveted over.
K = 2800 for stays screwed in and fitted with nuts.
K = 3000 for stays fitted with inside and outside nuts and washers, the outer washers being three times the diameter of the stay end and half the thickness of the plate.
K = 3400 for stays fitted with inside and outside nuts and washers, the outer washers having a diameter of ·6 × pitch, and a thickness of ·66 × thickness of plate, and being riveted to the plate.

If the combustion chamber top is supported by bridge stays only, which rest on the tube-plate, and are not assisted by any sling rods from the boiler shell, the thickness of the tube-plate must not be less than that given by the formula—

$$\text{Thickness} = \frac{P \times W \times p}{1800\,(p - d)}.$$

Wherein P = working pressure (kilos per square cm.).
W = width of chamber (tube-plate to back), in cm.
p = pitch of tubes, in cm.
d = internal diameter of plain tube, in cm.

If the tube-plates are of steel and are not exposed to the direct action of flame, the thickness may be reduced by 12½ per cent.

Section 5.—Combustion Chamber Bridge Stays.

The strength of these stays is to be that given by the following formula:—

$$\text{Thickness} = \frac{P\,(L - p)\,S \times L}{K \times d^2}.$$

Wherein P = working pressure (kilos. per square cm.).
L = length of bridge stay, in cm.
p = pitch of stay bolts in bridge stay, in cm.
S = distance apart of bridge stays, in cm.
d = depth of bridge stay at centre, in cm.
K = 420 for one stay bolt in each bridge stay.
K = 630 for two or three stay bolts.
K = 720 for four stay bolts.

"Thickness" stands for the collective thickness of the two plates of the bridge stay if constructed in that way.

If the plates are of steel they may be 10 per cent. thinner; and a further reduction may be made if they are assisted by slings from above.

GENERAL.

Manholes, &c., must be fitted with stiffening rings to compensate for plate cut away; they should not, as a rule, be less than 30 cm. × 40 cm., but in special cases 28 cm. × 38 cm. may be accepted. Cast-iron manhole covers cannot be accepted. It is strongly recommended that the studs of manhole covers be both screwed into the covers and riveted over on the inside.

Boilers are to be properly secured against movement both fore and aft and athwartships.

When there are several boilers they must be provided with proper shut-off valves to enable them to be worked independently.

Where the donkey boiler works at a lower pressure than the main boilers, satisfactory arrangements must be made to prevent steam passing from the latter into the former.

The space between the combustion chamber back and the back plate of the boiler should not be less than 12 cm., to allow for proper cleaning. In the cases of extra large boilers, and of boilers worked under forced draught, a greater space should be allowed.

Every marine boiler must be fitted with a blow-off valve and pipe, and also with a scum valve to draw the water from the level of the lower end of the water gauge. The blow-off pipe must be fitted with one cock on the boiler and a second on the ship's skin.

Wood must not be used either for lagging boilers or bulkheads in the neighbourhood of boilers.

All the larger mountings are to be secured to strong flanges riveted to the shell of the boiler; and the studs securing the mountings are not to penetrate the shell.

TESTS OF MATERIAL FOR STEEL BOILERS.

Regulations of 1*st July*, 1899.

All material to be of good and tough quality; any that proves brittle in working to be rejected.

A range of 5 kilos. will be allowed where ultimate tensile strength is 42 kilos. or under, and 6 kilos. where ultimate tensile is above 42 kilos.

Shell plates and other material not exposed to flame must show an elongation of at least 20 per cent. in a length of 200 millimetres; and material exposed to flame must show at least 22·5 per cent. on same length.

Material of less than 38 kilos. ultimate strength must elongate at least 25 per cent.

Bending tests are to be made on strips 25 millimetres wide, that have been heated to a dark red and quenched in water at 28° C.

They must stand bending without cracking to an angle of 180° over a radius, for shell plates, &c., of four times the thickness of the plate; and for material exposed to flame, of twice the thickness of the plate.

A tensile and a bending test is to be made from each shell plate. As regards other plates, a tensile test is to be made from every fourth plate; but, as far as possible, these test pieces should be so selected that there may be one from each furnace charge.

As regards plates other than shell plates and not exposed to flame, a bending test is to be made from each furnace charge. A bending test must be made from each plate that is exposed to flame.

Plates of over 41 kilos. tensile must be annealed if they have been worked in the fire.

Plates that are to be subjected to the direct action of flame should not have a higher ultimate tensile than 42 kilos. per square millimetre.

Steel of more than 46 kilos. ultimate tensile must not be used.

APPENDIX J.

BOARD OF TRADE RULES FOR SHAFTING.

SIZE OF SHAFTING.

MAIN, tunnel, propeller, and paddle shafts should not be passed, if less in diameter than that found by the following formulæ, without previously submitting the whole case to the Board of Trade for their consideration. It will be found that first-class makers generally put in larger shafts than those obtained by the formulæ.

For compound condensing engines with two or more cylinders, when the cranks are not overhung—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{f\left(2 + \frac{D^2}{d^2}\right)}}$$

$$P = \frac{f \times S^3}{C \times D^2}\left(2 + \frac{D^2}{d^2}\right).$$

Where S = diameter of shaft in inches.

d^2 = square of diameter of high-pressure cylinder in inches, or sum of squares of diameters when there are two or more high-pressure cylinders.

D^2 = square of diameter of low-pressure cylinder in inches, or sum of squares of diameters when there are two or more low-pressure cylinders.

P = *absolute* pressure in lbs. per square inch—that is, boiler pressure plus 15 lbs.

C = length of crank in inches.

f = constant from following table.

For ordinary condensing engines with one, two, or more cylinders, when the cranks are not overhung—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{3 \times f}}.$$

$$P = \frac{3 \times f \times S^3}{C \times D^2}.$$

Where S = diameter of shaft in inches.

D^2 = square of diameter of cylinder in inches, or sum of squares of diameters when there are two or more cylinders.

P = *absolute* pressure in lbs. per square inch—that is, boiler pressure plus 15 lbs.

C = length of crank in inches.

f = constant from following table.

TABLE LXXIII.

For Two Cranks. Angle between Cranks.	For Crank and Thrust Shafts.		For Tunnel Shafts.	For Propeller Shafts.
90°	For paddle engines of the direct-acting type, multiply constant in this column suitable for angle of cranks by 1·4.	1047	1221	890
100°		966	1128	821
110°		904	1055	768
120°		855	997	727
130°		817	953	694
140°		788	919	670
150°		766	894	651
160°		751	877	638
170°		743	867	631
180°		740	864	629
For Three Cranks. 120°	. . . .	1100	1295	943

When there is only one crank, the constants applicable are those in the table opposite 180°.

The portion of the propeller shaft which is forward of the stern gland, and all the thrust shaft, with the exception of the part enclosed in the thrust bearing, may be of the same diameter as the intermediate tunnel shafting.

LLOYD'S RULES FOR DETERMINING SIZES OF SHAFTS.

The diameters of intermediate shafts are to be not less than those given by the following formula :—

For compound engines, with two cranks at right angles—
Diameter of intermediate shaft in inches

$$= (\cdot04\,A + \cdot006\,D + \cdot02\,S) \times \sqrt{P}.$$

For triple-expansion engines, with three cranks at equal angles—
Diameter of intermediate shaft in inches

$$= (\cdot038\,A + \cdot009\,B + \cdot002\,D + \cdot0165\,S) \times \sqrt[3]{P}.$$

For quadruple-expansion engines, with two cranks at right angles—
Diameter of intermediate shaft in inches

$$= (\cdot034\,A + \cdot011\,B + \cdot004\,C + \cdot0014\,D + \cdot016\,S) \times \sqrt[3]{P}.$$

For quadruple-expansion engines with three cranks—
Diameter of intermediate shaft in inches

$$= (\cdot028\,A + \cdot014\,B + \cdot006\,C + \cdot0017\,D + \cdot015\,S) \times \sqrt[3]{P}.$$

For quadruple-expansion engines with four cranks—
Diameter of intermediate shaft in inches

$$= (\cdot033\,A + \cdot01\,B + \cdot004\,C + \cdot0013\,D + \cdot0155\,S) \times \sqrt[3]{P}.$$

Where A = diameter of high-pressure cylinder in inches.
B = diameter of first intermediate cylinder in inches.
C = diameter of second intermediate cylinder in inches.
D = diameter of low-pressure cylinder in inches.
S = stroke of pistons in inches.
P = boiler pressure above atmosphere in lbs. per square inch.

The diameter of crank-shaft, and of thrust-shaft under the collars, to be at least $\frac{21}{20}$ of that of the intermediate shaft. The diameter of thrust-shaft may be tapered off at each end to the same size as that of the intermediate shaft.

The diameter of the screw-shaft to be equal to the diameter of intermediate shaft (found as above) multiplied by $\left(\cdot 63 + \frac{\cdot 03\,P}{T}\right)$, but in no case to be less than 1·07 T, where P is the diameter of propeller, and T the diameter of the intermediate shaft, both in inches.

This size of screw-shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube. If no liners are used, or if two separate liners are used, the diameter of the shaft should be $\frac{21}{20}$ that given above.

The diameter of the screw-shaft is to be tapered off at the forward end to the size of the crank-shaft.

NOTE.—The rules are intended to apply to two cylinder compound engines, in which the ratio of areas of low- and high-pressure cylinders does not exceed 4·5 to 1; to triple-expansion engines in which it does not exceed 9 to 1; to quadruple-expansion engines in which it does not exceed 12 to 1; and in all cases, as regards the stroke, in which the length of stroke is not less than one-half the diameter or greater than the diameter of the low-pressure cylinder. Engines of extreme proportions beyond these limits being specially submitted to be dealt with on their merits.

BRITISH CORPORATION RULES FOR SHAFTING.

DIAMETER OF SHAFTING.

The minimum diameters of crank, thrust, propeller, and intermediate shafts may be found from the following formulæ, except where the ratio of length of stroke to distance between main bearings is unusual, when they will receive special consideration :—

$$D = \sqrt[3]{\frac{P \times L^2 \times S}{C}} \times B.$$

Where D = diameter of shaft.
P = absolute pressure—*i.e.*, boiler pressure + 15 lbs.
S = stroke of engine in inches.
L = diameter of low-pressure cylinder in inches.
B = 1·0 for crank and thrust shafts.
B = 0·95 for intermediate shafts.
B = for propeller shafts to be taken from the following table :—

TABLE LXXIV.

Coefficient of Displacement of Vessel at ⅘ Moulded Depth.	Ratio of Diameter of Propeller to Diameter of Crank-shaft.					
	13.	14.	15.	16.	17.	18.
·60	1·00	1·01	1·02	1·03	1·04	1·05
·62	1·01	1·02	1·03	1·04	1·05	1·06
·64	1·02	1·03	1·04	1·05	1·06	1·07
·66	1·03	1·04	1·05	1·06	1·07	1·08
·68	1·04	1·05	1·06	1·07	1·08	1·09
·70	1·05	1·06	1·07	1·08	1·09	1·10
·72	1·06	1·07	1·08	1·09	1·10	1·11
·74	1·07	1·08	1·09	1·10	1·11	1·12
·76	1·08	1·09	1·10	1·11	1·12	1·13
·78	1·09	1·10	1·11	1·12	1·13	1·14
·80	1·10	1·11	1·12	1·13	1·14	1·15

The value of the divisor, O, in the formula depends on the ratio $\frac{L^2}{H^2}$, where L = diameter of low-pressure cylinder, and H of high-pressure cylinder, in inches:—

TABLE LXXV.

$\frac{L^2}{H^2}$	Two Cranks at 90°, Compound or Quadruple, also Three Cranks at 120°, Quadruple-Expansion.	Three Cranks at 120°, Triple-Expansion.	Four Cranks at 90°, Quadruple-Expansion.
Ratio 3	9,910	...	...
,, 3⅛	10,160	...	...
,, 3¼	10,410	...	...
,, 3⅜	10,660	...	...
,, 3½	10,910	...	...
,, 3⅝	11,160	...	...
,, 3¾	11,410	...	...
,, 3⅞	11,660	...	...
,, 4	11,910	...	...
,, 4⅛	12,160	...	...
,, 4¼	12,410	...	...
,, 4⅜	12,660	...	...
,, 4½	12,910	13,650	...
,, 4¾	13,375	14,160	...
,, 5	13,840	14,670	...
,, 5¼	14,305	15,180	...
,, 5½	14,770	15,690	...
,, 5¾	15,235	16,200	...
,, 6	15,700	16,710	...
,, 6½	16,630	17,730	...
,, 7	17,560	18,630	...
,, 7½	18,410	19,530	...
,, 8	19,260	20,430	22,660
,, 8½	20,110	21,330	23,660
,, 9	20,960	22,200	24,660
,, 9½	21,750	23,070	25,660
,, 10	22,540	23,940	26,580
,, 10½	23,330	24,810	27,500
,, 11	24,120	25,660	28,420
,, 11½	24,900	26,500	29,340
,, 12	25,680	27,340	30,260

Intermediate ratios to have intermediate divisors.

Thrust and propeller shafts may be gradually tapered off to the diameter of the intermediate shafting.

The webs of built crank-shafts should be keyed as well as shrunk on to the shaft, and the diameter of the shaft should be increased in way of the web to make up the loss of sectional area at the key-way, and efficient dowel pins should be fitted in the crank pins.

Propeller shafts must be forged from scrap iron, rolled iron bars, or a single steel ingot made by the "open hearth" process. The tensile strength of iron shafts to be from 20 to 24 tons, with an elongation of 20 per cent. in 2 inches, and samples 1 inch square must stand being bent cold, without fracture, through 90° over a radius of 1½ inches. Steel shafts to have a tensile strength of from 27 to 35 tons per square inch, with an elongation varying from 33 per cent. with 27 tons to 25 per cent. with 35 tons tensile strength, measured on a gauge length of 2 inches on test pieces $\frac{9}{16}$ inch diameter. Samples 1 inch square must stand being bent cold, without fracture, to 180°. Tests are to be taken from one end of each shaft, and from both ends of shafts over 2½ tons weight, and of all propeller shafts. The test blocks are not to be removed until stamped by the surveyor, and test pieces are to be prepared therefrom as directed by him, and tested in his presence. Other material, such as nickel steel, &c., will be accepted after compliance with such special tests as may be imposed. All shafts are to be subjected to examination during the progress of manufacture, when rough turned and when finished.

BUREAU VERITAS RULES FOR SHAFTING.

A. CRANK SHAFTS.

When the crank of a screw engine is not overhung, the diameter of the shaft shall be determined by one of the following formulæ:—

For non-compound condensing engines—

$$d = \sqrt[3]{\frac{n\,P\,L\,D^2}{C}}. \quad \text{- - - - - - (A)}$$

For double, triple, and quadruple-expansion engines—

$$d = \sqrt[3]{\frac{P\,L\,(n_1\,D_1^2 + 0{\cdot}1\,n\,D^2)}{C}}. \quad \text{- - - (B)}$$

For shafts having a single overhung crank, the form under the radical sign is to be multiplied by

$$s + \sqrt{s^2 + 1}.$$

For two cylinder single crank tandem engines, the formula will therefore be—

$$d = \sqrt[3]{\frac{P\,L\,(D_1^2 + 0{\cdot}1\,D^2)\,(s + \sqrt{s^2 + 1})}{C}}. \quad \text{- (C)}$$

In these formulæ—

d = diameter of the after-shaft bearing in inches.

n_1 = number of high-pressure cylinders.

D_1 = diameter of each high-pressure cylinder in inches. If there are several high-pressure cylinders the diameters of which are not the same, $n_1 D^2_1$ represents the sum of the squares of their respective diameters.

n = number of low-pressure cylinders.

D = diameter of each low-pressure cylinder in inches. If there are several low-pressure cylinders the diameters of which are not the same, $n D^2$ represents the sum of the squares of their respective diameters.

N.B.—For triple or quadruple expansion engines, the intermediate cylinders do not come into account in the formulæ.

L = length of stroke in inches, common to all pistons.

P = boiler pressure above atmosphere in lbs. per square inch.

$s = \frac{a}{r}$ (see below). In order to determine a, B is supposed to be situated half-way the length of the bearing, unless the latter be longer than $1\frac{1}{8}$ times the diameter; in this case, B C may be considered as being equal to $\frac{3}{4}$ of the diameter.

C = a constant, the values of which for iron shafts are given below for certain cases. The values given apply to navigation in a seaway; for smooth water (tugs excepted) the constants may be increased by 30 per cent.

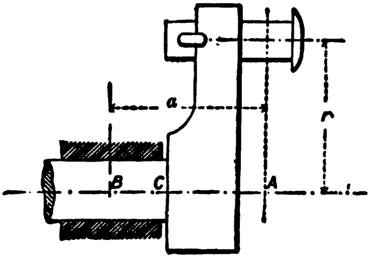

Fig. *k.*

If it is above 15 inches, it should be increased by an amount to be determined by the Administration; for built-up shafts, however, this latter increase will not be required.

For hollow shafts, the diameter must be increased by

1 per cent., if the diameter of the hole is 0·4 of the outside diameter.
2 " " " 0·5 "
5 " " " 0·6 "
10 " " " 0·7 "

If the hole is under 0·4 of the outside diameter, no increase will be required.

The Administration may allow a reduction on the diameter in certain special cases; for instance, in well-balanced engines with light moving parts, or for very superior workmanship, &c. On the other hand, the Administration may require an augmentation for engines which differ much from the average proportions found in practice; thus, for instance, for engines having a comparatively small stroke; for compound engines, the low-pressure cylinder of which has a very large size compared with the high-pressure cylinder, &c.

VALUES OF C IN FORMULÆ (A), (B), (C).

I. *Non-Compound Condensing Engines with Two Cylinders and Two Cranks.*

Formula (A), where $n = 1$ and $C = 4800$ when the angle between the cranks is 90°. When it is not 90° the constants are to be multiplied by the following coefficients:—

For 100°, - - -	0·91	For 140°, - - -	0·72
120°, - - -	0·79	160°-180°, - -	0·70

II. *Double-Expansion Engines (Ordinary Compound).*

1. Two-crank two-cylinder intermediate receiver engines.

Formula (B), where $n_1 = n = 1$.
$C = 3400$ when the angle between the cranks is 90°.

For other angles the constant is to be multiplied by the coefficients given in Case I.

2. Two-crank double tandem engines (two high- and two low-pressure cylinders).

Formula (B), where $n_1 = n = 2$.
$C = 3500$ when the angle between the cranks is 90°.

For other angles the constant is to be multiplied by the coefficients given in Case I.

3. Three-crank triple tandem engines (three high- and three low-pressure cylinders).

Formula (B), where $n_1 = n = 3$.
$C = 3800$ when the cranks are at 120°.

4. Single-crank tandem engines (one high- and one low-pressure cylinder). When the crank is *not* overhung—

Formula (B), where $n_1 = n = 1$.
$C = 2100$.

When the crank *is* overhung—

Formula (C), where $C = 3200$.

These two values suppose a cut-off at 0·8 of the stroke in the high-pressure cylinder. For later cut-offs, the constant is to be increased.

5. Three-crank three-cylinder engines (one high- and two low-pressure cylinders).

Formula (B), where $n_1 = 1$, $n = 2$.
$C = 3600$ when the cranks are at 120°.

III. *Triple-Expansion Engines.*

1. Three-crank three-cylinder engines (one high, one intermediate, and one low-pressure cylinder).

Formula (B), where $n_1 = 1$, $n = 1$.
$C = 3900$ when the cranks are at 120°.

2. Two-crank three-cylinder engines (two of the cylinders placed tandem-wise).

Formula (B), where $n_1 = n = 1$.
$C = 3000$ when the cranks are at right angles.

3. Two-crank four-cylinder engines (two tandem engines—viz., two high-pressure cylinders on the top of the intermediate and low-pressure cylinders).

Formula (B), where $n_1 = 2$, $n = 1$.
$C = 3300$ when the cranks are at right angles.

IV. *Other Cases.*

For quadruple-expansion engines, as well as for other cases not included in the foregoing, the constant will be determined by the Administration.

N.B.—For quadruple-expansion double tandem engines with two cranks at 90°, the value of C may be provisionally taken as 3100.

In the case of four cranks at right angles, the value of C may be taken at 3600. If the sequence of cylinders and the angles of the cranks are chosen so as to reduce the maximum torsional moment, the above value may be increased, but in no case shall exceed 4100. Each particular case to be submitted for approval.

B. PROPELLER, TUNNEL, AND THRUST SHAFTS.

The diameter of propeller shaft must be $\left(1{\cdot}7\frac{D}{d} - 15\right)$ per cent. in excess of the diameter of the crank-shaft, calculated from one of the formulæ, (A), (B), or (C), D being the diameter of the propeller in inches, and d the diameter of the crank-shaft in inches. It is recommended to fit the propeller shaft in such a way that it cannot move endways, if, for some reason or other, it has been uncoupled from the rest of the shafting. Liners fitted on propeller shafts to be tapered off at ends.

For tunnel shafts, a reduction of 6 per cent. on the diameter of the crank-shaft will be allowed.

The diameter of thrust-shaft at the bottom of the collars, both between and immediately beyond these latter, to be equal to that of the crank-shaft, and tapered off at each end to the smaller diameter of the body of the shaft. The thrust of the screw propeller must be taken up by an efficient thrust block, so as to prevent any fore and aft strain on the crank-shaft.

IRON SHAFTS FOR PADDLE STEAMERS.

In side-wheel steamers having double, triple, or quadruple-expansion engines with an intermediate shaft, each end of which carries an overhung crank-pin fitting loosely into an eye of the paddle-shaft crank, the bearing of the latter (see A of the following sketch) must have its diameter calculated from the formula:—

$$d = \sqrt[3]{\frac{P\,L\,(n_1 D_1^2 + 0{\cdot}1\,n\,D^2)\,(s + \sqrt{s^2 + 1})}{C}} \qquad \text{(D)}$$

where the letters have the same meaning as before, except that a, for determining $s = \frac{a}{r}$, is to be measured as shown in the following sketch, the point, B, being the middle of the bearing.

For two-cylinder compound receiver engines, with two cranks at 90°,

$$C = 13{,}000 \text{ for navigation in smooth water.}$$

For coasting or sea-going steamers, the constant is to be reduced, but need not be under 5700.

The diameter of the outer bearing of the paddle-shaft and of the intermediate shaft to be submitted to the Administration or the Surveyors for approval. The same applies to other cases not dealt with in this paragraph.

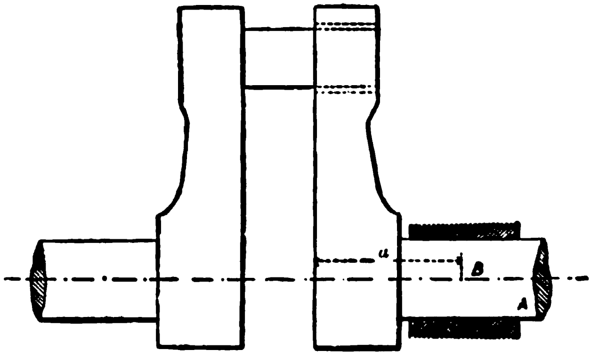

Fig. 1.

STEEL SHAFTS.

Shafts made of steel must have a tensile strength of 26 to 30½ tons, and test pieces cut from the forging must satisfactorily withstand the tests. When it is desired to make use of steel of softer or harder quality, the corresponding alteration in the diameter must be submitted to the Administration for approval.

APPENDIX K.

LLOYD'S RULES FOR THE USE OF ELECTRIC LIGHT ON BOARD VESSELS.

THE following requirements as to the sizes, positions, and protection of the cables, and to the fitting of the cut-outs are now embodied as rules:—

LEADS OR CIRCUITS.

1. The sectional area of the copper wires in the cables should be at least in the proportion of 1 square inch per 1000 amperes carried.

2. No single wire of greater size than 14 or of less than 18 standard wire gauge should be used. For portable leads, cables composed of stranded wires should be used, having sufficient conductivity and flexibility for the purposes intended.

3. The copper used in all wires or cables should have a conductivity of at least 98 per cent. that of pure copper.

4. The insulation resistance of all wires, including portable leads, should be not less than 600 megohms per statute mile, after twenty-four hours' immersion in sea water.

5. The insulating material used must not appreciably soften if subjected to a temperature of 180° F. If india-rubber insulation is used, the wires should be first covered with a layer of pure rubber, then with a separator, then with a layer of vulcanising india-rubber, and then with a layer of india-rubber-coated tape. The whole should then be vulcanised together. The cable should afterwards be satisfactorily protected, preferably with a braided covering of waterproof fibre.

6. Wires which are insulated with any other material than india-rubber should fulfil the same conditions as to insulation resistance, and should be of equal durability with those above specified.

JOINTS.

7. Joints in branches, or of branches with leads of small circuits, must be made in properly constructed watertight junction boxes, or should have the copper wires thoroughly soldered and the insulation carefully carried out, all the joints being made watertight. Joints in flow and return wires should not be made opposite one another. All joints should be in accessible positions, none being made in bunkers, cargo spaces, or spaces which may at any time be used for carrying cargo, stores, or baggage.

8. For soldering wires, resin only should be used as a flux.

9. Where practicable, the leads should be placed where they can always be accessible; if they are laid in wood battens, the covers should be screwed on, not nailed, and care should be taken that the casings are so arranged that water will not lodge in them. Cables which are properly covered with protective metal sheathing, or which are protected by galvanised wire armouring, may be unencased. They should, however, be secured by screwed clips, not by staples. All sharp bends in cables should be avoided.

10. All cables which are liable to be exposed to the weather or moisture should be lead covered, or be otherwise specially protected. Where great heat is experienced, no wood casing should be used, but the cables should be protected by iron casings, or, if they are not exposed to mechanical injury, they may be armoured with galvanised wire and fastened to decks or bulkheads with screwed clips spaced not more than 12 inches apart.

11. If cables are led through cargo spaces, coal bunkers, or spaces which may at any time be used for carrying cargo, stores, or baggage, or which are not at all times accessible, they should be strongly protected against damage, preferably by iron casings. If they are led through metal tubes, these must be strongly secured, and should be fitted so that water cannot lodge in them.

Armoured cables may be used without casings or tubes provided they are strongly secured to the underside of decks or to bulkheads by screwed clips, and provided they are armoured in conformity with the standard of the Engineering Standards Committee, viz. :—

For cables below ½ inch diameter over lead, by galvanised steel wires ·072 diameter; for cables ½ inch to 1 inch over lead, by two layers of steel tape each ·03 inch thick; for cables above 1 inch to 2 inches diameter, by two layers of steel tape each ·04 inch thick; and for larger cables, by two layers of steel tape ·06 inch thick.

12. Where cables pass through beams, bulkheads, or other iron work, they should be led through special fittings of sheet lead, hard wood, or vulcanised fibre to prevent their being chafed, and where they pass through decks they should be led through metal tubes lined with wood, or vulcanised fibre, and securely fastened to the decks, standing at such a height above the deck-level that water cannot stand above them. Where cables pass through watertight bulkheads, the fittings should be provided with brass watertight screwed glands.

13. In vessels having spaces allotted alternately for passengers and cargo, the lamp fittings in these spaces should be removable, and the terminals so arranged that they can be properly covered up with strong metal covers, or the whole of the fittings should be similarly provided with strong metal covers. The main switches and cut-outs should be outside these spaces, or, if placed inside, they should be in strong iron boxes provided with iron covers, or otherwise securely arranged to prevent the fittings being tampered with.

DISTRIBUTION.

14. A main switchboard should be fitted in the dynamo-room, to which all the main circuits throughout the ship should be brought, a switch and cut-out being fitted thereon for each circuit. The auxiliary switchboards for further subdivision of the current should be placed in conveniently accessible positions, and each such switchboard should be similarly fitted with a separate switch and cut-out for each sub-circuit. Cut-outs should be fitted to each lamp-circuit where these are made with reduced size of wire. If vessels are wired on the double-wire system, cut-outs should be fitted to each cable of these circuits.

15. In cases where electric lights are used for the mast-head light and side lights, the switches controlling these lights should be placed in a position where they can be controlled by the officer of the watch, or other

responsible person, and cannot be tampered with by other members of the crew, or by passengers, &c.

16. The switchboards should be of slate or other incombustible material. The switches should be on the quick-break principle, and should be so constructed that they must either be full "on" or completely "off"—that is, they must not be able to remain in an intermediate position. They should have ample rubbing surfaces, and their conductivity should not be less than that of the wires connected to them.

17. Cut-outs should be fitted to each main or auxiliary circuit, on the switchboards, as near as possible to the switches of these circuits. If the switchboard is not fitted near the dynamo, or if more than one dynamo may be used on any one circuit, then cut-outs should also be fitted to the main cable as near as possible to each of the dynamo terminals.

18. All other cut-outs should also be in easily accessible places, and as near as possible to the commencement of the cables or wires they protect. They should be mounted on slate or other incombustible bases, and be arranged so that the fused metal may not be a source of danger, and, where fitted with covers, these should be incombustible.

19. All fuses should be of easily fusible and non-oxidisable metal, and should be so proportioned as to melt with a current 100 per cent. in excess of the normal current—that is, they should melt with a current in the proportion of 2000 amperes per square inch of section of the wires they protect. The fuses for branch wires to single lights should, however, if of tin wire, be of not greater size than 22 S.W.G.

20. The fuses for each cable should be made of standard dimensions, so that a large fuse cannot be used for a small cable by mistake, or, if wire fuses are used, permanent instructions should be fitted on or near each switchboard giving particulars of the proper size of fuse for each circuit.

21. In shaft passages and in damp places, all lamp switches and cut-outs should be of a strong watertight pattern, or should be placed in watertight boxes having hinged or portable watertight covers. No switches or cut-outs are to be placed in bunkers.

22. There should be no joints in the cables leading from the dynamo to the main switchboard, nor in those leading from the main to auxiliary switchboards, nor should branches to single lamps be taken off these cables.

23. A voltmeter should be supplied with each installation. If more than one dynamo is fitted, neither being capable of the whole of the output, an ampere meter should be supplied with each dynamo.

JOINTS WITH HULL.

24. In vessels fitted on the single-wire system, all the joints with the hull should be placed in accessible positions. Those for single lamps or for small cables should be made with brass screws not less than three-eighths of an inch in diameter, carefully tapped into the iron or steel, having white brass washers, between the wires and the vessel, or the wires should be soldered to brass-faced washers. For larger cables and for the pole of dynamo, the cable wires should be properly sweated into brass or copper shoes, which should be bolted to the vessel. The iron or steel, where contact is made, should be filed bright, and the area of contact should not be less than eight times the section of the copper of the cable.

IN VESSELS CARRYING PETROLEUM.

25. The single-wire system must not be adopted for any part of the installation. Switches and cut-outs must not be fitted in places liable to the accumulation of petroleum vapour or gas, and all lamps in places where it is possible for gas to accumulate must be made with an outer glass globe made air-tight. All wires in such places are to be lead covered, or the insulation of the cables employed is to be of such a nature as not to be affected by petroleum. No joints of cables, switches, or cut-outs should be fitted in the pump-room, but the wires for each lamp therein should be carried to the lamp from a distributing junction box placed outside the pump-room or companion.

The following paragraphs, referring to the effect of the electric light installations upon the compasses, are issued as suggestions, not as rules :—

POSITION OF DYNAMOS AND OF ELECTRIC MOTORS.

26. The position and type of dynamos and electric motors should be such that the compasses will not be affected. Dynamos and large motors should be at least 30 feet from the standard compass.

CABLES.

27. In vessels fitted with continuous current dynamos and wired on the single-wire system, no single cable should be carried within 15 feet of any compass, and cables conveying heavy currents should be fixed at still greater distance. If it is necessary to fix any cables within this distance, then, for all parts of the vessel lighted from this cable, the concentric or double-wire system should be adopted, the return wire being carried as near the flow as possible in the vicinity of the compasses.

ADJUSTMENT OF COMPASSES.

28. The compasses should be adjusted with the dynamo not working, after which the vessel's head should be put upon the different courses, with the dynamo running at full speed, and on each course the indications of the compass should be noted with the dynamo running with open circuit, and with all possible combinations of the current switched "on" and "off" all circuits passing near the compasses. These indications should be compared with those obtained with the dynamo stopped, and any serious deflections of the compasses remedied before the vessel sails.

The requirements in paragraphs 3 to 8 inclusive, referring to the quality of the material used or to the workmanship employed, are embodied as rules; but, as the quality of the material can only be tested at the cable-maker's works, and as the workmanship of the joints cannot be examined or tested after completion, the guarantee of the electrical engineers will be required as to these points.

APPENDIX L.

Board of Trade Rules for Safety-valves.

Provisions of the Act as regards Safety-valves.—Every steamship of which a survey is required by the Act shall be provided with a safety-valve upon each boiler, so constructed as to be out of the control of the engineer when the steam is up, and, if such valve is in addition to the ordinary valve, it shall be so constructed as to have an area not less, and a pressure not greater, than the area of and pressure on that valve.

The surveyors are instructed that, in all *new boilers* and whenever *alterations can be easily made*, the valve chest should be placed directly on the boiler; and the neck, or part between the chest and the flange which is bolted on to the boiler, should be as short as possible and be cast in one with the chest.

If any person place an undue weight on the safety-valve of *any* steamship, or, in the case of steamships surveyed under the Act, increase such weight beyond the limits fixed by the engineer-surveyor, he shall, in addition to any other liability he may incur by so doing, be liable for each offence to a fine not exceeding one hundred pounds.

The engineer-surveyor shall declare, amongst other things, the limits of the weight to be placed on the safety-valves; that the safety-valves are such, and in such condition, as required by the Act; and that the machinery is sufficient for the service for the time he fixes, and is in good condition.

Area of Safety-valves.—When natural draught is used, the area per square foot of fire-grate surface of the locked-up safety-valves should not be less than that given in the following table opposite the boiler pressure intended, but in no case should the valves be less than 2 inches in diameter. This applies to new vessels and to vessels which have not previously received a passenger certificate.

When, however, the valves are of the common description, and are made in accordance with the table, it will be necessary to fit them with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits.

When forced draught is used, the area of the safety-valves should not be less than that found by the following formula:—

$$\frac{A \times C}{20} = \text{area of valves required.}$$

Where A = area of valves as found from the table.
C = estimated coal consumption in lbs. per sq. ft. of grate per hour.

TABLE LXXVI.—Safety-valve Areas.

(NATURAL DRAUGHT.)

Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.
Lbs.	Sq. inch.	Lbs.	Sq. inch.	Lbs.	Sq. inch.	Lbs.	Sq. inch.
15	1·250	67	·457	119	·279	170	·202
16	1·209	68	·451	120	·277	171	·201
17	1·171	69	·446	121	·275	172	·200
18	1·136	70	·441	122	·273	173	·199
19	1·102	71	·436	123	·271	174	·198
20	1·071	72	·431	124	·269	175	·197
21	1·041	73	·426	125	·267	176	·196
22	1·013	74	·421	126	·265	177	·195
23	·986	75	·416	127	·264	178	·194
24	·961	76	·412	128	·262	179	·193
25	·937	77	·407	129	·260	180	·192
26	·914	78	·403	130	·258	181	·191
27	·892	79	·398	131	·256	182	·190
28	·872	80	·394	132	·255	183	·189
29	·852	81	·390	133	·253	184	·188
30	·833	82	·386	134	·251	185	·187
31	·815	83	·382	135	·250	186	·186
32	·797	84	·378	136	·248	187	·185
33	·781	85	·375	137	·246	180	·184
34	·765	86	·371	138	·245	189	·183
35	·750	87	·367	139	·243	190	·182
36	·735	88	·364	140	·241	191	·181
37	·721	89	·360	141	·240	192	·181
38	·707	90	·357	142	·238	193	·180
39	·694	91	·353	143	·237	194	·179
40	·681	92	·350	144	·235	195	·178
41	·669	93	·347	145	·234	196	·177
42	·657	94	·344	146	·232	197	·176
43	·646	95	·340	147	·231	198	·176
44	·635	96	·337	148	·230	199	·175
45	·625	97	·334	149	·228	200	·174
46	·614	98	·331	150	·227	201	·173
47	·604	99	·328	151	·225	202	·173
48	·595	100	·326	152	·224	203	·172
49	·585	101	·323	153	·223	204	·171
50	·576	102	·320	154	·221	205	·170
51	·568	103	·317	155	·220	206	·169
52	·559	104	·315	156	·219	207	·169
53	·551	105	·312	157	·218	208	·168
54	·543	106	·309	158	·216	209	·167
55	·535	107	·307	159	·215	210	·166
56	·528	108	·304	160	·214	211	·166
57	·520	109	·302	161	·213	212	·165
58	·513	110	·300	162	·211	213	·164
59	·506	111	·297	163	·210	214	·164
60	·500	112	·295	164	·209	215	·163
61	·493	113	·292	165	·208	216	·162
62	·487	114	·290	166	·207	217	·161
63	·480	115	·288	167	·206	218	·161
64	·474	116	·286	168	·204	219	·160
65	·468	117	·284	169	·203	220	·159
66	·462	118	·281				

When the pressure exceeds 180 lbs. per square inch, the accumulation of pressure at the steam test will probably be exceptionally high, unless the area of the branch leading from the valve chest is in excess of the area of the valves, and the area of the main waste steam pipe is correspondingly in excess of the gross area of the valves.

In ascertaining the fire-grate area, the length of the grate should be measured from the inner edge of the dead plate to the front of the bridge, and the width from side to side of the furnace on the top of the bars at the middle of their length.

Examination of Safety-valves.—The surveyor, in his examination of the machinery and boilers, is particularly to direct his attention to the safety-valves, and, whenever he considers it necessary, he is to satisfy himself as to the pressure on the boiler by actual trial.

The surveyor is to examine the whole of the valves, weights, and springs at every survey.

The responsibility of seeing to the efficiency of the mode by which the valves are fitted, so as to be out of the control of the engineer when steam is up, rests with the surveyor, who should see that the method adopted is efficient and approved of by the Board of Trade.

The safety-valves should be fitted with lifting gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting gear should, in all cases, be arranged so that it can be worked by hand either from the engine-room or stokehole.

Care should be taken that the safety-valves have a lift equal to at least one-fourth their diameter; that the openings for the passage of steam to and from the valves, including the waste steam pipe, should each have an area not less than the area of valves required by Table lxxvi.; that each valve-box has a drain-pipe fitted at its lowest part. In the case of lever valves, if the holes in the lever are not bushed with brass, the pins must be of brass; iron and iron working together must not be passed. The valve seats should be secured by studs and nuts.

Surveyor to See Valves Weighted.—When the surveyor has determined the amount of pressure, he is to see the valves weighted accordingly, and the weights or springs fixed in such a manner as to preclude the possibility of their shifting or in any way increasing the pressure. The limit of the weight on the valves is to be inserted in the declaration, and should it at any time come to a surveyor's knowledge that the weights have been shifted or the loading of the valves otherwise altered, or that the valves have been in any way interfered with, so as to increase the pressure, without the sanction of the Board of Trade, he is at once to report the facts to the Board of Trade.

Spring Safety-valves.—If the following conditions are complied with, the surveyor need raise no question as to the substitution of spring loaded valves for dead-weighted valves:—

(1) That at least two valves are fitted to each boiler.
(2) That the valves are of the proper size, as required by Table lxxvi.
(3) That the springs and valves be so cased in that they cannot be tampered with.
(4) That provision be made to prevent the valves flying off in case of the springs breaking.

(5) That the requisite safety-valve area is cased and locked up in the usual manner of the Government valves.

(6) That screw lifting gear be provided to meet all the valves.

(7) That the size of the steel of which the springs are made is to be found by the following formula:—

$$\sqrt[3]{\frac{s \times D}{c}} = d.$$

s = the load on the spring in lbs.
D = the diameter of the spring (from centre to centre of wire) in inches.
d = the diameter, or side of square, of the wire in inches.
c = 8000 for round steel.
c = 11,000 for square steel.

(8) That the springs be protected from the steam and impurities issuing from the valves.

(9) That, when valves are loaded by direct springs, the compressing screws abut against metal stops or washers when the loads sanctioned by the surveyor are on the valves.

(10) That the springs have a sufficient number of coils to allow a compression under the working load of at least one-quarter the diameter of the valve.

The size of steel of springs of safety-valves should not, as a rule, be less than ¼ inch.

In no case is the surveyor to give a declaration for spring-loaded valves unless he has tried them under full steam and full firing for at least twenty minutes with the feed-water shut off and stop-valve closed, and is fully satisfied with the result of the test. In special cases, or when the valves are of novel design, the results of the test under full steam should be reported to the Board, but, if the surveyor is fully satisfied with them, he need not delay the granting of the declaration for the vessel pending the approval of the Board. If, however, the accumulation of pressure exceed 10 per cent. of the loaded pressure, he should withhold his declaration, and report the case to the Board of Trade, accompanied by a sketch, if necessary, and stating the strength pressure and the working pressure of the boilers.

In the case of safety-valves, of which the principle and details have already been passed by the Board of Trade, the surveyor need not require plans to be submitted, so long as the details are unaltered, of which he must fully satisfy himself; but in any new arrangement of valves, or in any case in which any detail of approved valves is altered, he should, before assuming the responsibility of passing them, report particulars, with a drawing to scale, to the Board of Trade.

The tracings of new safety-valve designs should, if possible, be transmitted to the Board of Trade for consideration before the construction of the safety-valves is commenced.

Owners, Masters, and Engineers to see that Valves are kept in Proper Order.—It is clearly the duty of the masters and engineers of vessels to see, in the intervals between the surveys, that the locked-up safety-valves, as well as the other safety-valves and the rest of the machinery, are in proper working order. There is no provision in the Merchant Shipping

Act, 1854, exempting the owner of any vessel, on the ground that she has been surveyed by the Board of Trade surveyors, from any liability, civil or criminal, to which he would otherwise be subject. The Act of Parliament requires the Government safety-valve to be out of the control of the engineer when the steam is up; this enactment, far from implying that he is not to have access to it, and to see to its working, at proper intervals when the vessel is in port, rather implies the contrary; and the master should take care that the engineer has access to them for that purpose. Substantial locks that cannot be easily tampered with, and as far as possible weather-proof, should be used for locking up the safety-valve boxes.

All Tests for Pressure and Accumulation are to be made with Board of Trade Gauges.—In witnessing the hydraulic tests of boilers, &c., and in witnessing all safety-valve tests for accumulation of pressure, the surveyors are to use the pressure-gauges supplied by the Board of Trade for the purpose. No steam gauge should be used without having a syphon filled with water between it and the boiler.

APPENDIX M.

Admiralty Rules for Testing Materials for Machinery.

(e) *Steel Castings for Machinery.*

Steel castings. Tests

Steel castings for the machinery are to satisfy the following conditions:—Tensile strength, not less than 28 tons per square inch, with an extension of 2 inches of length of at least 23 per cent. Bars of the same metal, 1 inch square, should be capable of bending cold, without fracture, over a radius not greater than 1⅝ inches, through an angle depending on the ultimate tensile strength, this angle to be not less than 90° at 28 tons ultimate strength, and not less than 60° at 35 tons ultimate strength, and in proportion for strengths between these limits. For intricate thin castings the extension in 2 inches of length is to be at least 10 per cent., and the bending angle is to be not less than 20° at 28 tons ultimate strength, and 15° at 35 tons ultimate strength, and in proportion for strengths between these limits. Test pieces are to be taken from each casting. All steel castings are also to satisfactorily stand a falling test, the articles being dropped from a height of 12 feet (or as may be approved) on a hard macadamised road or a floor of equivalent hardness.

It is to be distinctly understood that contractions or defects in steel castings are not to be made good by patching, burning, or by electric welding, without the sanction of the Admiralty overseers.

(f) *Steel Forgings for Machinery.*

Steel forgings.

All steel forgings are to satisfy the following conditions:—Ultimate tensile strength, not less than 28 tons per square inch, with an extension in 2 inches of length of at least 30 per cent. Bars of the same metal, 1 inch square, should be capable of being bent cold, without fracture, through an angle of 180° over a radius not greater than ¼ inch. Test pieces are to be taken from each forging. Crank and propeller shafts are to have test pieces taken from each end, and the ultimate tensile strength of the material of these shafts must not exceed 32 tons per square inch.

For all important forgings, such as crank and propeller shafts, connecting- and piston-rods, the forgings are to be gradually and uniformly forged from solid ingots, from which at least 30 per cent. of the top end of the ingot has been removed before forging, and at least 3 per cent. of the total weight of the ingot from the bottom end after forging. The sectional area of the body of the finished forging is to be not more than ⅙ the original sectional area of the ingot.

(*g*) *Cast Iron.*

Cast iron.

Test pieces to be taken from such castings as may be considered necessary by the inspecting officer. The minimum tensile strength to be 9 tons per square inch, taken on a length of not less than 2 inches. The transverse breaking load for a bar 1 inch square, loaded at the middle between supports 1 foot apart, is not to be less than 2000 lbs.

(*h*) *Gun-metal, Naval Brass, and White Metal.*

Composition, analysis, and tests of gun-metal and naval brass, &c.

The gun-metal used for all castings throughout the whole of the work supplied by the contractors is, unless otherwise specified, to contain not less than 8 per cent. of tin, and not more than 5 per cent. of zinc, the remainder to be of approved quality copper. The exact proportion of tin above 8 per cent. being arranged as may be required, depending on the use for which the gun-metal is intended. The ultimate tensile strength of gun-metal is to be not less than 14 tons per square inch, with an extension in 2 inches of length of at least 7½ per cent. The composition of any naval brass used is to be:—Copper, 62 per cent.; zinc, 37 per cent.; and tin, 1 per cent. All naval brass bars are to be cleaned and straightened. They are to be capable of (1) being hammered hot to a fine point, (2) being bent cold through an angle of 75° over a radius equal to the diameter or thickness of the bars. The ultimate tensile strength of naval brass bars ¾ inch diameter and under is not to be less than 26 tons per square inch, and for round bars above ¾ inch diameter, and square bars not less than 22 tons per square inch, whether turned down in the middle or not. The extension in 2 or 4 inches of length is to be at least 10 per cent. Breaks within less than ½ inch of the grip are not to count. Cuttings from the propellers and other important gun-metal castings and naval brass work will be sent to Portsmouth dockyard for analysis. The white metal used for bearing surfaces is to contain at least 85 per cent. of tin, not less than 8 per cent. of antimony, and about 5 per cent. of copper; zinc or lead should not be used. The brasses are to be carefully tinned before filling with white metal.

Castings and forgings.

The gun-metal castings are to be perfectly sound, clean, and free from blowholes. The steel castings are required to be clean, sound, and to be out of twist, and as free as possible from blowholes; the steel forgings are required to be quite sound, clean, and free from all flaws. All castings and forgings must admit of being machined, planed, and bored to the required dimensions; and no piecing, patching, bushing, stopping, or lining, will be permitted, nor will any manufactures in which these conditions have been infringed be accepted. In cases of doubt as to the suitability of castings or other materials, for the purpose intended, early reference should be made to the inspecting officer to avoid subsequent delay by the rejection of such parts after delivery. The whole of the steel plates, angles, and rivets used in the construction of any part of the work supplied is to be manufactured by the Siemens-

Martin process. All castings are to be of steel or gun-metal, except where otherwise specified; and all forgings, plates, bolts, &c., are to be of steel. No steel is to be toughened without sanction from the Admiralty.

Copper for Pipes.

Copper.

Strips cut from the steam and other pipes, either longitudinally or transversely, are to have an ultimate tensile strength of not less than 13 tons per square inch when annealed in water, with an elongation in length of 2 inches or 4 inches of not less than 35 and 30 per cent. respectively. Such strips are also to stand bending through 180° cold until the two sides meet, and of hammering to a fine edge without cracking.

Treatment of Mild Steel.

Hammer scale to be removed.

All plates and furnaces for boilers and steam pipes are to be treated as follows, with a view of removing the black oxide or scale formed during manufacture:—The plates and furnaces, previous to their being taken in hand for working, are to stand for not less than 8 hours in a liquid consisting of 19 parts of water and 1 of hydrochloric acid. The plates should be placed in the bath on edge, and not laid flat. When the plates and furnaces are removed from the dilute acid both the surfaces are to be well brushed and washed to remove any scale which may still adhere to them. They should then be placed in another similar bath filled and kept well supplied with fresh water, or be thoroughly washed with a hose, as may be found necessary. The plates on removal from the fresh water should be placed on edge to dry. This treatment is to be carried out on the premises where the boilers and pipes are made.

Cold bending.

All plates or bars which can be bent *cold* are to be so treated; and if the whole length cannot be bent cold, heating is to be had recourse to over as little length as possible.

Hydraulic flanging.

The front and end plates of the boilers, and also all other plates, including those for the combustion chambers, *are to be flanged by hydraulic pressure*, and in as few heats as possible.

Dangerous temperature to be avoided.

In cases where plates or bars have to be heated, the greatest care should be taken to prevent any work being done upon the material after it has fallen to the dangerous limit of temperature known as a "blue heat"—say from 600° to 400° Fahrenheit. Should this limit be reached during working, the plates or bars should be reheated.

Annealing.

Plates or bars which have been worked while hot are to be subsequently annealed simultaneously over the whole of each plate or bar.

Failure of metal.

In cases where any bar or plate shows signs of failure or fracture in working it is to be rejected. Any doubtful cases are to be referred to the Admiralty.

APPENDIX N.

TURBINES.

THE steam turbine is now no longer in the experimental stage so far as marine propulsion is concerned, inasmuch as ships of all sizes in both the Naval and Mercantile Services have been tried with such motors, with results sufficiently satisfactory, it would seem, to warrant the placing of orders for more ships; and, so far as the Admiralty is concerned, to the exclusion of the reciprocating engine. It remains, however, to be seen if such a course is justified by the test of time; for, after all, mere speed is not the only thing of importance in warships of every sort, or even in all kinds of passenger steamers. The small screws and methods of reversing them must always make the ships having them somewhat unhandy, and, in a battleship, manœuvring powers seem to be of even higher importance than speed. Moreover, with the larger powers, the difficulty of dealing with the huge and heavy parts of the turbine and its case preclude such repairs at sea as are possible, and have been accomplished, with the reciprocating engine; and these repairs will be difficult in most foreign ports and stations where crane-power is limited. In the case of a breakdown of a comparatively small nature, stoppage is a necessity; whereas, with the old engine, it is not always so, and, generally, some sort of a repair was possible by the engine-room staff to enable the reciprocators to bring the ship into port. The design and manufacture of the turbine is an art to itself, and cannot be dealt with satisfactorily in such a work as this; moreover, it has been done so fully and admirably by Mr. A. Jude,* whose book may be studied to great advantage by all interested in the subject of marine propulsion. But, for a summary of the subject, perhaps one cannot do better than read what Mr. Frank Foster has said, viz.:—

"The turbine itself is naturally a very high-speed machine, and the screw propeller a relatively slow-speed machine. A compromise has to be made—the turbine speed being reduced while that of the propeller is increased. For instance, if the blades of a large marine turbine were to have a peripheral speed of 300 feet per second—which is often exceeded in land practice—and the screw propeller were to retain its normal speed of, say, 90 revolutions per minute, the diameter of the rotor would be 64 feet. Needless to say, this is impossibly big, and in practice the blade speed would be cut down to about one-third this value, while the revolutions per minute would be doubled or trebled. This reduction in the normal peripheral and rotational speeds of the turbine profoundly influences both turbine and propeller. It is very nearly true that the number of stages required varies inversely as the square of the blade speed, so that reducing the mean effective blade speed by 50 per cent. will quadruple the number of stages. Also, the diameter of rotor necessary to attain a given blade

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* *The Theory of the Steam Turbine*, by A. Jude. (Charles Griffin & Co., London, 1906.)

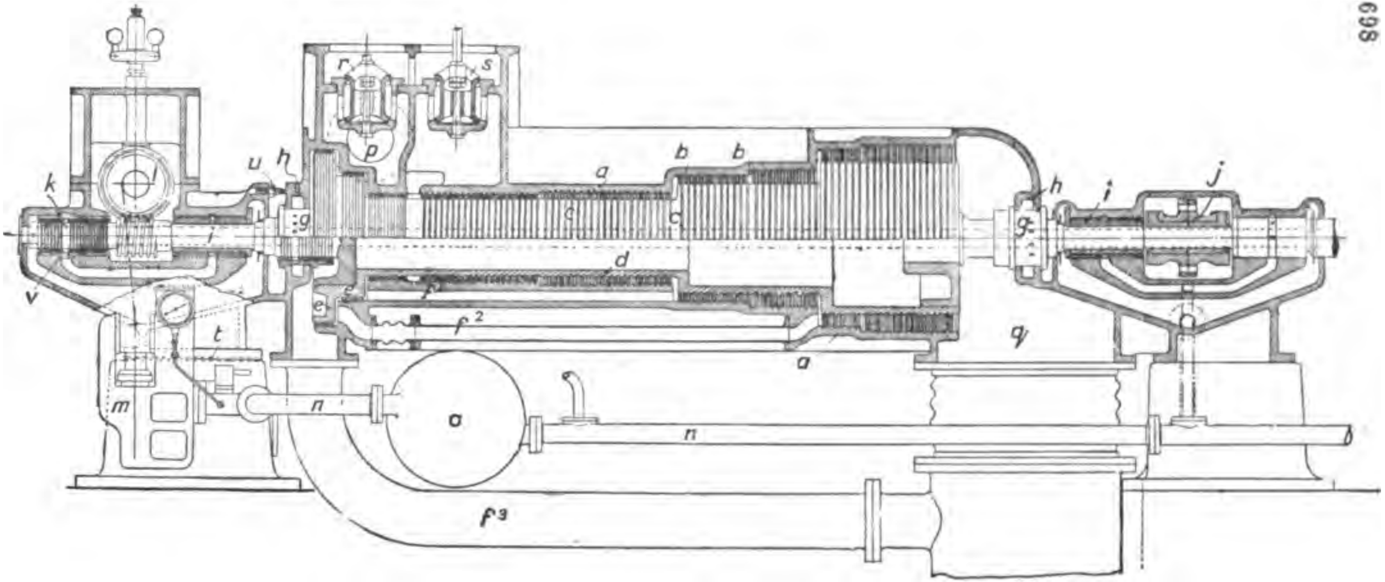

Fig. m.—Section of Parsons Turbine.

a, Casing in halves.
b, Fixed vanes.
c, Moving vanes.
d, Rotating drum.
e, Dummy or balance pistons.
f^1, *f*^{2}, *f*^{3}, Pressure equilibrium passages.
g, Glands.
h, Steam to glands.
i, Flexible bearings.
j, Flexible coupling.
k, Thrust or longitudinal adjusting block.
l, Worm gearing for driving oil pump and governors through bevel gearing.
m, Oil pump.
n, Oil supply pipe—oil return on other side.
o, Oil cooler.
p, Steam inlet.
q, Steam exhaust.
r, Emergency valve.
s, Governor valve.
t, Feather to keep bearings, &c., central while casing expands.
u, Screw for adjusting top half of thrust.
v, Liners for adjusting bottom half of thrust.

speed is inversely proportional to the revolutions. These two factors combine to make the turbine bulky, heavy, and expensive. In most marine turbines there is no considerable saving in weight or cost as compared with reciprocators. Floor space is not saved, although head room is, and the attendance required is often somewhat reduced. The influence on the propeller of the increased rotational speed is considerable. The size of the propeller is reduced, and, in order to partially overcome this difficulty, three propellers are commonly used, with the blades very considerably widened. Two or three propellers have been placed on the same shaft with the object of increasing the propeller area, but this cannot have any great influence for good. In spite of these changes, the combined propeller disc areas are generally too small, with the result that the slip must be great.

"The screw propeller is a fluid-momentum engine. It influences or controls a certain weight of water, giving to it a certain sternward velocity, and thus imparting to the water a certain quantity of momentum. The forward thrust on the ship is precisely equal to this momentum. Now, it is usually considered that all the water passing through the propeller circle is acted upon by the propeller, and hence the smaller this circle, or circles, the greater must be the water velocity across it; for we know that the momentum imparted to the water is strictly proportional to the product of the quantity of water passing per second and the velocity given to it. Consequently, reducing the area of these circles requires an increase in the sternward velocity of the water—that is, in the slip. The effect of increasing the blade widths is to enable them to more effectively control the water passing across the propeller circle, and therefore, in reality, to increase the effective quantity of water passing. This, however, is only useful up to a certain point, and there is the increase in the friction of the propeller to be remembered. This propeller friction is also increased by the small pitch ratio which has to be adopted. The result is that the friction loss and the kinetic energy thrown into the wake are both high and the propeller efficiency correspondingly low.

"As a set-off against this, we have the fact that the mechanical efficiency of the turbine and its propeller shafting is considerably greater than that of a corresponding reciprocating engine, due to the greater simplicity of the turbine itself, to the practical balance of the end thrust without the use of a large thrust block, and to the smaller size of shafting required. The adoption of steam turbines is limited by questions of economy and cost. In slow or low-powered ships, the diameter or length, or both, of the turbines is bound to be great and the turbine expensive. If the diameter is as large as is necessary in order to secure a reasonable blade speed, the blade length becomes very small and the internal leakage of steam over the blade tips relatively large, with, consequently, a considerable reduction in economy. This last objection applies mainly to 'Parsons'—reaction—turbines and not to impulse turbines, as in the latter case partial peripheral admission of the steam from the fixed blades is possible, and there is no appreciable clearance leakage over the blade tips. The former objection—that of bulk and cost—applies to impulse turbines, and it is doubtful if turbines can be successfully applied to any type of ship of less than 13 or 14 knots speed, and for all except the very largest ships the limit is probably nearer 16 or 17 knots." Professor Rateau places the limit still higher.

"The preceding generalisation should be qualified to some extent. The use of a low-pressure turbine taking steam at rather less than atmospheric pressure from a reciprocating engine would be quite practicable for comparatively small-powered and slow-speed ships. The combination would be exceptionally economical—quite 10 per cent. better than the existing reciprocating engines—and would further ensure ample manœuvring power without the employment of astern turbines. At present most turbine boats lack manœuvring power, a quality of most importance where the runs are short, as in cross-channel and river service. Undoubtedly the best conditions for successful turbine propulsion are high speed, great power, and calm weather. As usually fitted on board ship, there are three 'Parsons' turbines on separate shafts. The high-pressure turbine is on the central shaft, and exhausts into two low-pressure turbines in parallel—thermally—on the wing shafts. These wing shafts also carry astern turbines placed at the exhaust ends of the main turbines, so that when the ship is going ahead the turbines are revolving in the condenser vacuum and offer very little resistance."

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